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Distributed Mixing Burner (DMB) Engineering Design for Application to Industrial and Utility Boilers

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16. ABSTRACT The report summarizes the design of two prototype distributed mixing burners (DMBs) for application to industrial and utility boilers. The DMB is a low-NOx pulverized-coal-fired burner in which: (1) mixing of the coal with combustion air is controlled to minimize NOx emissions, and (2) an overall oxidizing environment is maintained to avoid slagging and corrosion. Several DMB configurations were tested in two research furnaces over a range of operating conditions. The data were evaluated to develop design criteria for optimum performance. Two prototype DMBs were then designed by integrating the design criteria with commercial burner components. One burner was designed for application to an industrial size (215,000 lb/hr) Foster Wheeler boiler. The second prototype burner was designed for application to a general class of Babcock and Wilcox opposed-fired utility size boilers, because a utility-size host boiler had not been selected.

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TABLE OF CONTENTS

Page

1.0 SUMMARY 1-1

2.0 INTRODUCTION 2-1

3.0 PROTOTYPE DISTRIBUTED MIXING BURNER DESIGNS 3-1

3.1 Host Site Requirements 3-1
3.1.1 Initial Industrial Boiler 3-1
3.1.2 Utility Boiler 3-4

3.2 DMB Subsystem Design Specificiations 3-7
3.2.1 Fuel System Variables 3-12
3.2.2 Secondary Air Variables 3-17
3.2.3 Tertiary Air Variables 3-23
3.2.4 Burner Exit Geometry 3-28
3.2.5 General DMB Design Requirements 3-31

3.3 Commercial Burner Practice 3-33
3.3.1 Standard Burners 3-33
3.3.2 Low NO_x Burners 3-43

3.4 Comparison of General DMB Design Requirements with Standard Commercial Practice 3-44

3.5 Integrated Prototype Burner Designs 3-46
3.5.1 Industrial Boiler Prototype Burner 3-47
3.5.2 Utility Boiler Prototype Burner 3-50

4.0 PREDICTION OF PROTOTYPE BURNER FIELD OPERATING CHARACTERISTICS 4-1

4.1 Foster Wheeler Intervane Burner (Pre-NSPS) 4-1
4.2 Babcock & Wilcox Dual Register Burner (Post-NSPS) 4-4
4.3 Projection of DMB NO_x Levels to Field Conditions 4-6

APPENDIX A--DEVELOPMENT DISTRIBUTED MIXING BURNER DESIGN A-1

APPENDIX B--RESEARCH FURNACE DESIGNS B-1

APPENDIX C--EFFECTS OF OPERATING VARIABLES AND FUEL COMPOSITION ON DMB PERFORMANCE C-1

APPENDIX D--LOW NO_x BURNER COMMERCIAL DESIGNS AND PERFORMANCE D-1

APPENDIX E--COMMERCIAL BURNERS TESTED IN THE LWS E-1

APPENDIX F--ABBREVIATIONS F-1

APPENDIX G--METRIC CONVERSIONS G-1

LIST OF FIGURES

Figure	Title	Page
2-1	Distributed Mixing Burner (DMB) Concept	2-2
2-2	Distributed Mixing Burner	2-3
3-1	Cross Section of Initial Industrial Boiler	3-2
3-2	Burner Load Schedule for Initial Industrial Boiler	3-6
3-3	Burner Capacities for Babcock & Wilcox Pulverized Coal-Fired Boilers	3-8
3-4	Effect of Primary Stoichiometry on NO/CO Characteristics for Single Secondary Burner(s)	3-14
3-5	Effect of Primary Swirl Angle on NO and CO Using the 50 x10^6 Btu/hr Single Secondary Burner	3-16
3-6	Effect of Primary Nozzle Setback on NO_x Emissions (12.5 x 10^6 Btu/hr Single Secondary Burner)	3-18
3-7	Effect of Primary Air Velocity on Operating Limits for 50 x 10^6 Btu/hr Dual Secondary Burner	3-19
3-8	Effect of Burner Zone Stoichiometry on NO and CO Emissions with Two Single Secondary - Short Flame Burners	3-21
3-9	Effect of Secondary Swirl Angle on NO and CO Characteristics (50 x 10^6 Btu/hr Single Secondary Burner	3-22
3-10	Effect of Burner Zone Stoichiometry on NO Characteristics for Long Flame Burners	3-24
3-11	Effect of Tertiary Port Location on NO and CO Characteristics	3-26
3-12	Effect of Tertiary Air Velocity on NO and CO Characteristics	3-27
3-13	Effect of Tertiary Port Location and Number on NO/CO Characteristics in a Multiburner Configuration	3-29
3-14	Effect of Quarl Geometry on NO Characteristics	3-30

LIST OF FIGURES (Cont'd)

Figure	Title	Page
3-15	Typical PC Burner Design	3-38
3-16	Foster Wheeler Intervane Burner	3-39
3-17	Typical Air Registers	3-40
3-18	Cross Section of Industrial Prototype Burner Based on Foster Wheeler Design, Without Tertiary Air Ports	3-49
3-19	Tertiary Control Valve	3-51
3-20	Tertiary Port Arrangement for Industrial Prototype Burner	3-52
3-21	Cross Section of Preliminary Utility Prototype Burner Based on Babcock and Wilcox Design	3-54
3-22	Coal Nozzle Designs	3-58
4-1	Performance/Emissions Characteristics for Foster Wheeler Intervane Burner	4-3
4-2	Performance/Emission Characteristics for Babcock/Wilcox DRB in LWS - Single Burner	4-5
4-3	Foster Wheeler NO_x Emission Correlation	4-7
4-4	Babcock and Wilcox Correlation	4-9
A-1	Distributed Fuel/Air Mixing Concept	A-3
A-2	Dual Secondary Throat Distributed Mixing Burner (50×10^6 Btu/hr)	A-5
A-3	Operating Map for 50×10^6 Btu/hr Dual Secondary Throat Burner with Utah Coal, Primary Stoichiometry 20% T.A. and Combustion air Temperature 600°F.	A-8
A-4	Swirl Number for Dual Secondary Throat Burner (50×10^6 Btu/hr)	A-10
A-5	Flame Characteristics with Divided Throat Burner	A-12
A-6	Single Secondary Burner Design	A-13
A-7	Operating Map for Burner Air Velocities of Various Loads and Overall Stoichiometry - 50×10^6 Btu/hr Single Secondary Burner Design	A-16

LIST OF FIGURES (Cont'd)

Figure	Title	Page
A-8	Swirl Characteristics for the Single Secondary Burner Design	A-17
A-9	Typical Flame Shape for Single Secondary Burner	A-19
B-1	Large Watertube Simulator (LWS) Test Facility	B-2
B-2	Variable Geometry LWS Combustor	B-3
B-3	Airflow Distribution	B-5
B-4	Small Watertube Simulator (SWS) Test Facility	B-6
B-5	Control and Data Acquisition Layout	B-8
B-6	Desk Computer Output	B-9
B-7	Overall Scheme of Measurement and Control of SWS/LWS Facilities	B-13
B-8	Flow Systems for Exhaust Gas Analyzers	B-15
B-9	Exhaust Gas Sample Conditioning System	B-16
C-1	Effect of Overall Stoichiometry on NO Characteristics for All Single Secondary - Short Flame Burners	C-3
C-2	CO Characteristics as a Function of Overall Stoichiometry for the Single Secondary - Short Flame Burners	C-4
C-3	Effect of Burner Heat Release on NO Characteristics for the Short Flame DMB Design	C-6
C-4	Effect of Excess Air on NO Characteristics - Long Flame Burner Designs	C-7
C-5	NO Characteristics with Differing Fuels for Short Flame Burners	C-9
C-6	Effect of Fuel Type on NO_x Emissions for the 10×10^6 Btu/hr Single Secondary - Long Flame Burner	C-11
C-7	Comparison of Operating Limits for Two 50×10^6 Btu/hr Burners Having Distinctly Differing Flame Characteristics	C-13
C-8	Operating Limits for a Single Large Capacity Burner and a Four Burner Array of Smaller Capacity Burners	C-14

LIST OF FIGURES (Cont'd)

Figure	Title	Page
D-1	Foster Wheeler Controlled Flow Burner	D-2
D-2	Effect of Secondary Air Balancing	D-3
D-3	Foster Wheeler Controlled Flow Burner Performance	D-5
D-4	Foster Wheeler Controlled Flow-Split Flame Low NO_x Burner	D-6
D-5	Foster Wheeler Low NO_x Burner Performance in Test Facility	D-8
D-6	Babcock & Wilcox Dual Register Pulverized Coal Burner	D-10
D-7	B&W Dual Register Burner Field Test Results	D-11
E-1	Foster Wheeler Intervane Burner	E-2
E-2	Babcock & Wilcox Burner (50×10^6 Btu/hr)	E-4

LIST OF TABLES

Table	Title	Page
3-1	Initial Industrial Boiler Demonstration Site Characteristics	3-3
3-2	Initial Industrial Boiler Demonstration Site Coal Characteristics	3-5
3-3	Approximate Utility Boiler Demonstration Site Characteristics	3-9
3-4	Development DMBs Tested in Research Furnaces	3-10
3-5	Prototype Burner Design Parameters	3-34
3-6	Coal Supply Characteristics for Pulverized Coal-Fired Burners	3-36
3-7	Comparison of Prototype Burner Design Criteria with Standard Commercial Practice	3-45
3-8	Industrial Prototype Burner Dimensions and Adjustments	3-53
3-9	Preliminary Utility Prototype Burner Dimensions	3-56
A-1	Design Information for Dual Secondary Burner	A-7
A-2	Design Information for Single Secondary Burners	A-15
B-1	General Measurement Scheme	B-12
B-2	Analytical Instruments for Combustion Gases	B-14
C-1	Composition of Coals Tested	C-8
D-1	Controlled Flow and Controlled Flow-Split Flame Test Results	D-7
E-1	Intervane Burner Characteristics	E-3
E-2	Babcock & Wilcox Burner Design Point Operation Specifications	E-5

1.0 SUMMARY

The "Distributed Mixing Burner"(DMB) is a low NO_x pulverized coal burner that provides for controlled mixing of the coal with combustion air to minimize NO_x emissions while maintaining an overall oxidizing environment in the furnace to minimize slagging, fouling and corrosion. DMBs have been tested at firing rates up to 100×10^6 Btu/hr in single and four burner arrays in two research furnaces under EPA Contracts 68-02-1488 and 68-02-2667. These tests covered wide ranges of burner parameters and operating conditions. The data were evaluated to develop design criteria for future DMBs.

Two prototype DMBs were designed for checkout testing in a research furnace followed by installation in field operating boilers. One burner was designed for application to an industrial size (215,000 lb/hr) Foster Wheeler Boiler under EPA Contract 68-02-3127. This boiler is owned and operated by Western Illinois Power Cooperative (WIPCO) in Pearl, Illinois. The DMB design criteria were integrated with commercial burner components by Foster Wheeler Energy Corporation, a subcontractor under this program. The second prototype burner was designed for application to a general class of Babcock and Wilcox opposed fired utility size boilers since a utility size host boiler had not been selected. The DMB design criteria were integrated with Babcock and Wilcox commercial burner components by Babcok and Wilcox under EPA Contract 68-02-3130.

This report discusses the initial prototype burner designs. Subsequent burner testing and burner design changes are discussed in the annual reports under EPA Contracts 68-02-3127 and 68-02-3130.

2.0 INTRODUCTION

The DMB concept involves staging the combustion process to minimize NO_x emissions while maintaining an overall oxidizing atmosphere in the furnace to minimize slagging and corrosion. NO_x production from fuel nitrogen compounds is minimized by driving a majority of the compounds into the gas phase under fuel-rich conditions and providing a stoichiometry/temperature history which maximizes the decay of the evolved nitrogen compounds to N_2. Thermal NO_x production is also minimized by enthalpy loss from the fuel-rich zone which reduces peak temperatures.

A schematic representation illustrating how the DMB design sequentially stages the fuel/air mixing is shown in Figure 2-1. The combustion process occurs in three zones. In the first zone pulverized coal transported by the primary air combines with the inner secondary air to form a very fuel-rich (20 to 50 percent theoretical air) recirculation zone which provides flame stability. The coal devolatilizes and fuel nitrogen compounds are released to the gas phase. Outer secondary air is added in the second "burner zone" where the stoichiometry increases up to about 70 percent theoretical air. This is the optimum range for reduction of bound nitrogen compounds to N_2. Air to complete the combustion process is supplied through tertiary ports located outside the burner throat. This allows substantial residence time in the burner zone for decay of bound nitrogen compounds to N_2 and radiative heat transfer to reduce peak temperatures. The tertiary ports surround the burner throat providing an overall oxidizing atmosphere and minimizing interactions between adjacent burners.

Figure 2-2 shows one configuration of the DMB as it would be installed in a typical wall-fired boiler. The DMB design components illustrated are:

- Four independently controlled air stream systems divide the total combustion air into:
  1. A central airstream used for ignition during start-up and cooling during load operation;
  2. The primary airstream carrying the coal;
  3. Two concentric secondary airstreams to provide inner and outer secondary airflow; and
  4. Four outboard tertiary air ports.

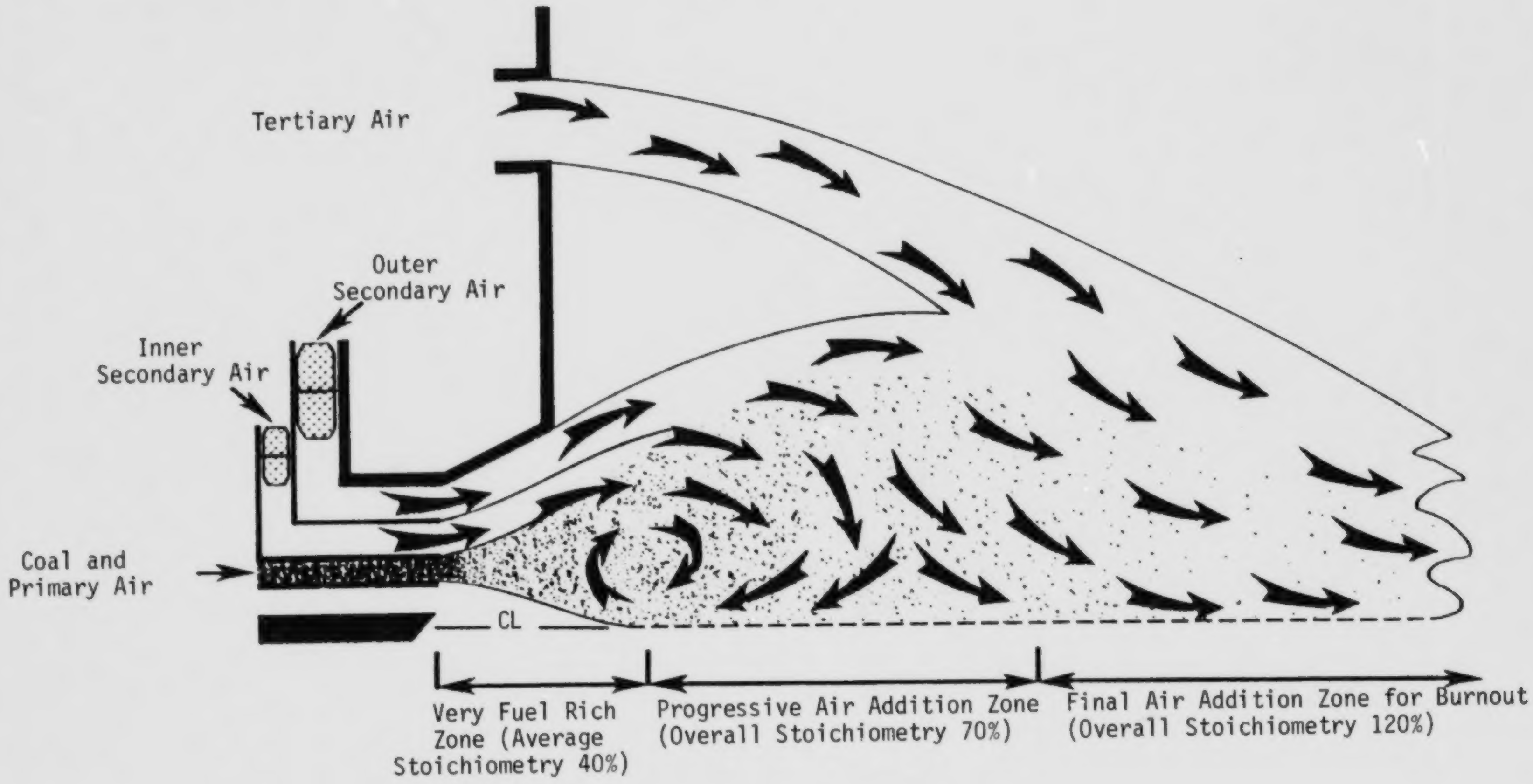

Tertiary Air
Outer Secondary Air
Inner Secondary Air
Coal and Primary Air
CL
Very Fuel Rich Zone (Average Stoichiometry 40%)
Progressive Air Addition Zone (Overall Stoichiometry 70%)
Final Air Addition Zone for Burnout (Overall Stoichiometry 120%)

Figure 2-1. Distributed Mixing Burner (DMB) Concept.

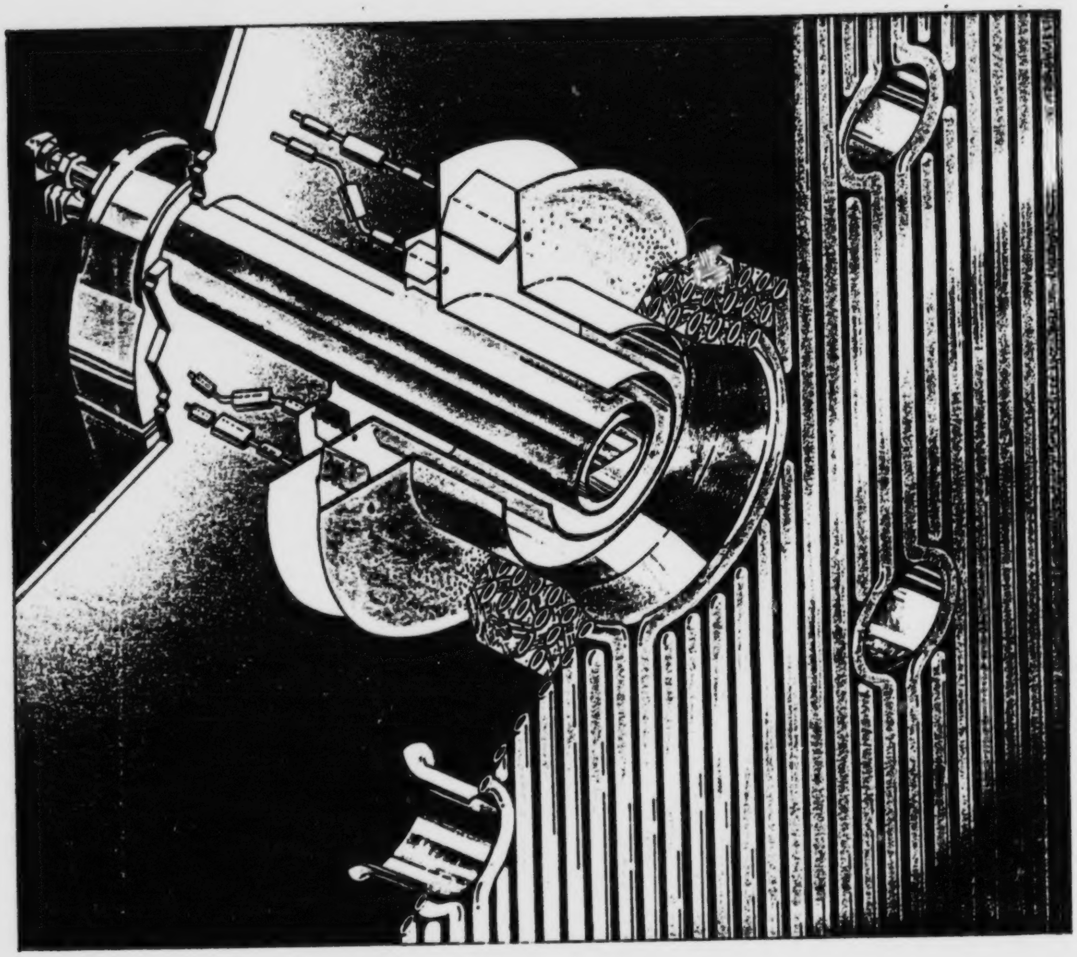


Figure 2-2. Distributed Mixing Burner.

- An annular fuel injector which produces a uniform annular coal distribution.
- Two adjustable air registers connected to the secondary air passages to control both the airflow rates and degree of swirl.
- A retractable oil gun on the burner axis.
- Ignition and main flame scanner systems for safety.

Alternate configurations using central coal nozzles and other burner component designs could also be used.

For the last several years, Energy and Environmental Research Corporation (EER) has been working with the EPA to develop the DMB concept under EPA Contracts 68-02-1488 and 68-02-2667. These efforts have focused on experimentally evaluating the performance of several DMB designs in two research furnaces. Six DMBs have been tested in single and four burner arrays at firing rates up to 100×10^6 Btu/hr. The tests covered wide ranges of DMB design parameters, adjustments and operating conditions. Under optimum conditions, NO_x levels less than 0.15 $lb/10^6$ Btu were achieved. However, DMB performance has not been evaluated in commercially operated steam generators.

The EPA is currently conducting two programs to provide this field evaluation. EPA Contract 68-02-3127 with EER as the prime contractor and Foster Wheeler as a major subcontractor involves evaluation of the DMB concept on two industrial size single wall fired boilers. EPA Contract 68-02-3130 with Babcock and Wilcox as the prime contractor and EER as a major subcontractor is an evaluation of the DMB concept on a utility size opposed fired boiler. The objectives of both programs are to achieve NO_x emission levels less than 0.2 $lb/10^6$ Btu without adverse effects on boiler operability and durability, thermal efficiency and the emission of other pollutants. These field evaluations involve: 1) translation of the development burner test data into practical prototype DMBs, 2) verification of the prototype burner performance through testing in a research furnace, 3) construction and installation of these burners in the field boilers, 4) evaluation of burner/boiler performance under typical operating conditions, and 5) documentation of the results.

This report discusses the first program element, the prototype DMBs. These burners will be tested in a research furnace at EER to fine tune the burner designs and to adjust operating variables for an optimum balance between low NO_x emissions and other performance parameters. Based on this testing, the final design will be specified and the necessary burners fabricated for installation in the host boilers. Since the EPA low NO_x delayed mixing burners will be installed in boilers designed to accommodate burners of conventional design, the retrofit burners must conform to current burner design practice with respect to fuel and air supplies (temperatures, pressures, coal size distribution, etc.), burner operating envelopes (turndown, excess air, furnace size, etc.) and burner performance (flame length, CO emissions, etc.).

Contemporary burners as currently designed are expected to:

- Ignite the coal in a stable manner and provide the proper flame shape over the entire load range with no or minimal adjustments to the burner;
- Provide for proper fuel/air mixing so that low unburned combustible losses are maintained and emissions levels meet local regulations;
- Operate continuously between scheduled annual outages without major parts replacement and meet accepted maintenance standards;
- Operate safely under all operating conditions.

Typical boiler and site specifications also can affect the ability of the burner to achieve the above performance, safety, emissions and maintenance requirements such as:

- Boiler configuration (burner zone heat release rate, furnace depth, firing configuration, capacity);
- Burner specifications (capacity, number of burners, spacing of burners, pulverizer); and
- Fuel (grade, ash properties, nitrogen and sulfur levels, moisture, etc.).

Fulfilling all of the performance, safety and emissions requirements necessitated the development of prototype DMB designs which meet three types of design requirements:

- Functional requirements of EPA low NO_x delayed mixing burner ;
- Contemporary boiler/burner design requirements, and;

- Specific requirements of host boilers

This report details the design approach for the prototype burners which relied heavily on an understanding of the basic processes/mechanisms controlling both emissions and efficiency. Due to the multitude of burner sizes, boiler configurations and fuels, a prime objective was to establish generalized design criteria. However, defining generalized optimum burner design criteria requires a clear understanding of the interaction of all the burner geometrical, flow and mixing parameters with the physical and chemical processes occurring within the boiler, as well as the impact of these processes on emissions and performance. The link between (1) the burner, (2) exhaust emissions, and (3) performance is through the mixing and combustion characteristics occurring within the boiler. However, due to the technical difficulty in obtaining in-flame measurements and the associated costs, generalized design criteria were not established and an empirical approach was taken for definition of the DMB design.

Even taking the empirical approach, the multiplicity of parameters precluded the establishment of generalized optimum design criteria, and therefore, the effort concentrated on establishing the trends in performance/emissions/stability characteristics with variations in burner input parameter, burner design parameters, tertiary air design, burner scale, multiburner effects and fuel impacts. In this manner a set of design specifications was established for a prototype burner design based on site-specific requirements. The key to this design is its flexibility, enabling it to be "tuned" for optimization in the research furnace. The final optimized configurations will be installed and evaluated in the field operating boilers.

The prototype burner designs discussed in this report are these initial "flexible" designs based on the results of previous DMB development efforts. In the course of testing the industrial prototype burner in the Large Watertube Simulator (LWS-EPA Contract 68-02-3127), considerable changes in some design parameters were made including the incorporation of some proprietary Foster Wheeler components and parameter values. Similarly, although the utility prototype burner (EPA Contract 68-02-3130) has not yet been tested, it is expected that the final burner design may be considerably different from the initial prototype burner design included in this report.

The following section details the prototype burner design process illustrating the integration of DMB design criteria (generated from development DMB tests) with host site characteristics and commercial burner components. Section 4.0 presents an approximate means of predicting the DMB performance in field operating boilers. The appendix includes: A) descriptions of the development DMB designs, B) the characteristics of the research furnaces used for the development DMB tests, C) additional development DMB test results, D) characteristics of commercial low NO_x burners, E) characteristics of commercial burners tested under these programs, and F) abbreviations.

3.0 PROTOTYPE DISTRIBUTED MIXING BURNER DESIGNS

The prototype burners have been designed to meet the following requirements:

- Compatibility with host boilers and ancillary equipment;
- Distributed mixing burner design criteria;
- Commercial burner components, construction methods and operational characteristics.

The prototype burners will serve as the design basis for field operating burners to be installed at specific host sites. Thus, they are being designed to be fully compatible with the host boilers' firing systems, furnaces, and ancillary equipment. Section 3.1 provides a description of the host site requirements, and Section 3.2 contains the distributed mixing burner subsystem design criteria. In Section 3.3, a description of commercial burner requirements that must be met with the DMB is also included, and in Section 3.4, the DMB design requirements are compared with commercial practice. This section ends with detailed descriptions of the prototype burner designs (Section 3.5).

3.1 Host Site Requirements

The initial site for the industrial boiler demonstration has been selected, and a prototype burner has been fully designed based on the host site specifications. The utility boiler demonstration site has not yet been selected. A preliminary prototype burner design has, however, been developed based on the general characteristics of utility boilers in the size range of interest.

3.1.1 Initial Industrial Boiler

The boiler selected as the initial industrial boiler demonstration site is Pearl Station Unit No. 1 which is owned and operated by Western Illinois Power Cooperative (WIPCO), Jacksonville, Illinois. This unit has a maximum continuous rating of 215,000 lb/hr of steam and is single wall-fired with four burners in a 2 x 2 array. Figure 3-1 is a cross-sectional view of the unit, and Table 3-1 lists the characteristics which impact prototype burner design.

The windbox depth of 52 inches is adequate for the currently installed Intervane burners which have a single register and a short exit. However, this depth is too short to accommodate the dual registers and long exit specified in the prototype design criteria. The maximum exit length which can be used is 12 inches. This corresponds to about 40 percent of the prototype

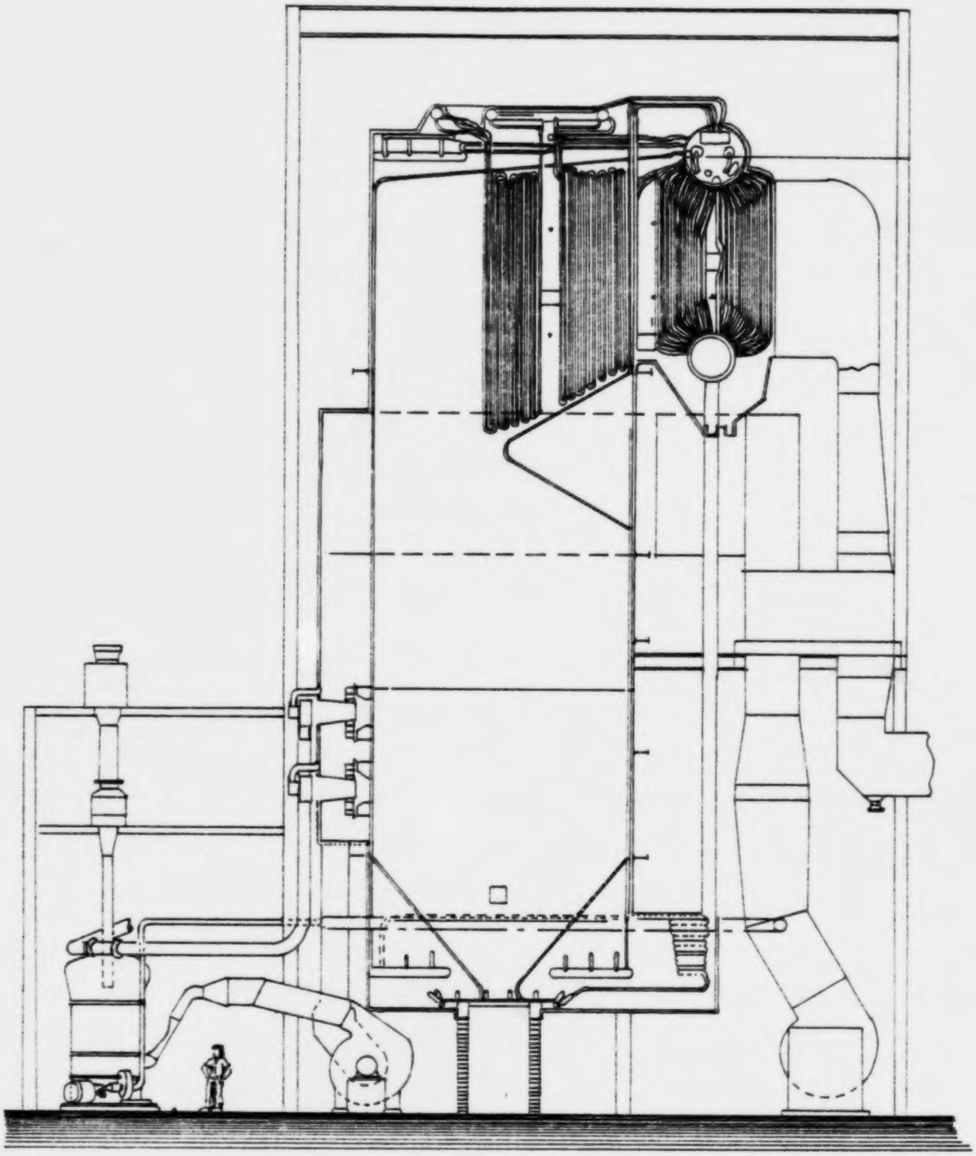


Figure 3-1. Cross Section of Initial Industrial Boiler.

Table 3-1. Initial Industrial Boiler Demonstration Site Characteristics

Characteristic		Value
Boiler/Furnace		
Boiler Capacity:	MCR*	215 x 10^3 lb/hr
	Peak	245 x 10^3 lb/hr
Furnace Depth		20.6 ft
Furnace Width		16.4 ft
Firing Configuration		Single Wall
Burner Capacity	MCR	69 x 10^6 Btu/hr
	Peak	80 x 10^6 Btu/hr
Current Burners	Type	Intervane
	Throat Diameter	24 in.
Burner Array, Wide x High		2 x 2
Burner Spacing:	Horizontal	6.0 ft
	Vertical	6.5 ft
Fuel Supply System		
Coal Type		Bituminous
Mills:	Type	MB
	Number	2
Primary Stoichiometry at:	MCR	16% T.A.**
Primary Temperature		155°F
Air Supply System		
Air Heater Type		Regenerative
Design Point Excess Air at:	MCR	18%
Windbox Temperature at:	MCR	510°F
Burner Pressure Drop (Normal)		3.5 in. H_2O
Windbox Depth		52 in.

*MCR = Maximum Continuous Rating

**T.A.= Theoretical Air

burner throat diameter (30.5 inches), whereas the design criteria specifies 100 percent. The furnace depth on this unit is typical of single wall-fired units and will require a short flame to avoid rear wall flame impingement. The coal currently supplied to this unit is an Indiana bituminous coal from the Three State Trucking Mine in Huntingburg. Table 3-2 lists the characteristics of this coal. In comparison with the Utah coal used in the majority of distributed mixing burner testing at EER, this coal has: higher moisture and ash, comparable volatiles and heating value, and lower nitrogen content. The coal is pulverized by two mills each feeding two burners. Under low load conditions one mill and two burners are removed from service and the air registers on the two burners out of service are closed.

Figure 3-2 shows the load schedule (i.e., the variation of operating conditions with load). The primary temperature and stoichiometry were measured with only one mill in service. However, the results are typical of two-mill operation as well. The primary airflow rate is almost constant, increasing about 15 percent as the load is increased from 50 to 80×10^6 Btu/hr per burner. This results in the decrease in primary stoichiometry as load increases. The primary temperature is usually maintained in the range of 150^0 to 160^0F, but in the mill test, it was allowed to increase to 175^0F.

The excess air levels for this unit are shown for two conditions: all burners in service at the boiler design point, and two burners in service. This unit uses spray attemperation to control steam temperature. An excess air level of 18 percent allows the design point steam temperature to be maintained down to about half of MCR. With two burners out of service, excess air is increased at low load to maintain steam temperature. The windbox temperature depends on the performance of the regenerative air heater which is related to the boiler load and excess air.

3.1.2 Utility Boiler

The utility boiler demonstration site has not been selected, and thus, it is not posssible at this time to precisely define the boiler characteristics which influence prototype burner design. However, the general characteristics of the family of boilers from which the field evaluation site will be selected can be identified and used as the basis for a preliminary prototype burner design.

Table 3-2. Initial Industrial Boiler Demonstration Site Coal Characteristics

	As Received	Dry Basis
Proximate Analysis		
% Moisture	13.03	
% Ash	10.14	11.66
% Volatile	36.08	41.49
% Fixed Carbon	40.75	46.85
	100.00	100.00
Heating Value, Btu/lb	11,216	12,896
% Sulfur	2.87	3.30
% Alk. as Na_2O	0.19	0.22
Sulfur Forms		
% Pyritic Sulfur	1.74	2.00
% Sulfate Sulfur	0.03	0.03
% Organic Sulfur	1.10	1.27
% Total Sulfur	2.87	3.30
Water Soluble Alkalies		
% Na_2O	0.006	0.007
% K_2O	0.003	0.004
	Reducing	Oxidizing
Fusion Temperature of Ash		
Initial Deformation	2080°F	2370°F
Softening (H = W)*	2140°F	2410°F
Softening (H = ½W)	2190°F	2450°F
H = cone height		
W = cone width		
Fluid	2320°F	2550°F

Hardgrove Grindability Index = 69.0
(HGI @ 9.47% moisture)

*** H = Height, W = Width**

	As Received	Dry Basis
Ultimate Analysis		
Moisture	13.03	
Carbon	61.30	70.48
Hydrogen	4.36	5.01
Nitrogen	1.30	1.49
Chlorine	0.03	0.04
Sulfur	2.87	3.30
Ash	10.14	11.66
Oxygen (diff)	6.97	8.02
	100.00	100.00

	% Weight Ignited Basis
Mineral Analysis	
Phos. pentoxide, P_2O_5	0.15
Silica, SiO_2	44.03
Ferric oxide, Fe_2O_3	26.85
Alumina, Al_2O_3	21.49
Titania, TiO_2	1.02
Lime, CaO	1.33
Magnesia, MgO	0.77
Sulfur trioxide, SO_3	0.50
Potassium oxide, K_2O	2.39
Sodium oxide, Na_2O	0.31
Undetermined	1.16
	100.00

Base: Acid Ratio = 0.48
Silica Value = 60.33
T250 = 2340°F

Estimated Viscosity at Critical Viscosity Temperature of 2390°F = 125 Poises

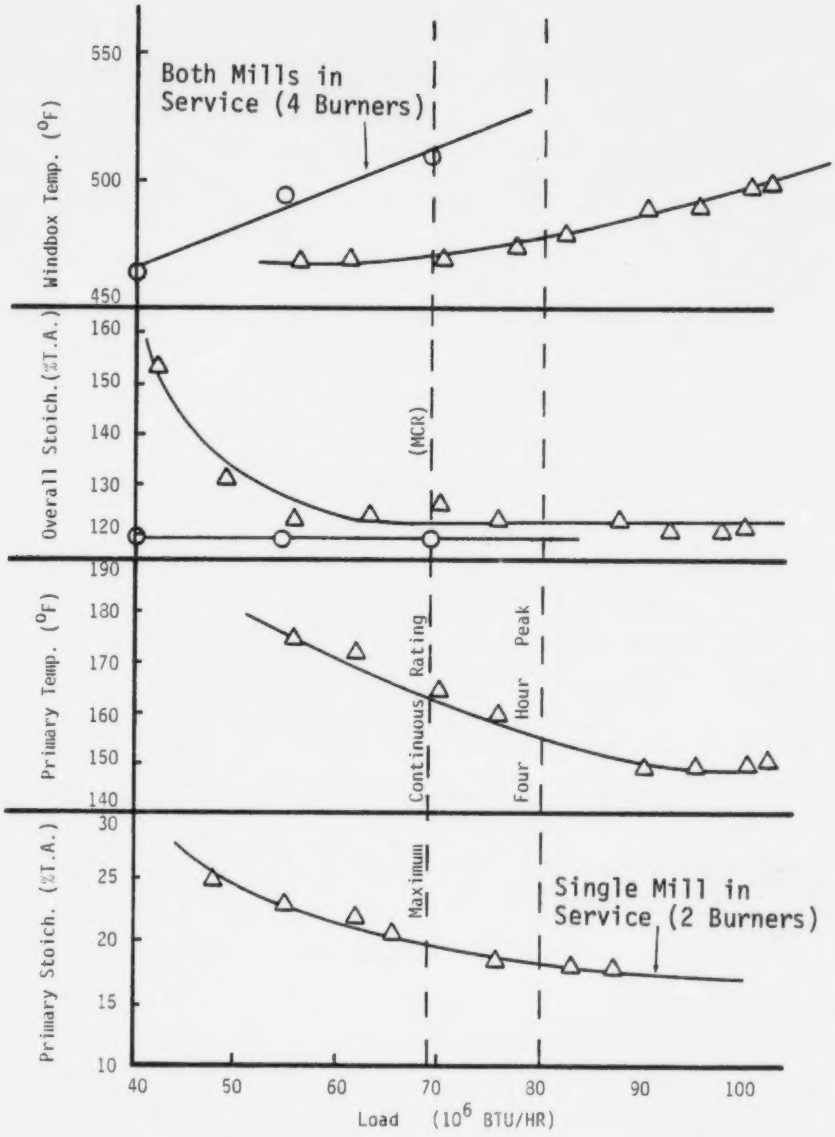


Figure 3-2. Burner Load Schedule for Initial Industrial Boiler.

The utility boiler field evaluation site will be a Babcock and Wilcox unit of about 350 MW capacity, and will probably be opposed wall-fired. There will be about 24 burners (12 per wall) arrayed in a 3 x 4 or 4 x 3 (wide x high) array of circular burners or a 2 x 2 array of cell burners (three closely space circular burners). Typical furnace depths for these units are 32 to 36 feet. This large furnace depth is sufficient to allow for considerably longer flames than with the single wall-fired industrial boiler field evaluation site. The burner capacities for Babcock and Wilcox pulverized coal-fired boilers erected since 1962 are shown in Figure 3-3. The average capacity is about 125×10^6 Btu/hr. This and other approximate characteristics of the utility boiler demonstration site are listed in Table 3-3.

3.2 DMB Subsystem Design Specifications

This section presents the results of development DMB tests in the research furnaces at EER leading to the specification of design criteria for the prototype DMBs. Six DMB configurations were tested in two research furnaces. The DMB designs are listed in Table 3-4 and are categorized as either long or short flame designs. The long flame designs utilized a central coal nozzle producing an axial coal and primary air jet which extended a considerable distance into the furnace. The resulting flames were long and narrow. The short flame designs also utilized a central coal nozzle but were equipped with an impeller to impart some swirl to the primary coal and air stream. As a result, the coal mixed more rapidly with the secondary and tertiary combustion air producing a shorter flame. The design details of these development DMBs are presented in Appendix A and the research furnaces are described in Appendix B.

The burner design variables can be organized into the following four categories.

- Fuel Variables
  - Primary Temperature
  - Primary Stoichiometry
  - Primary Velocity
  - Primary Swirl Angle
- Secondary Air Variables
  - Temperature

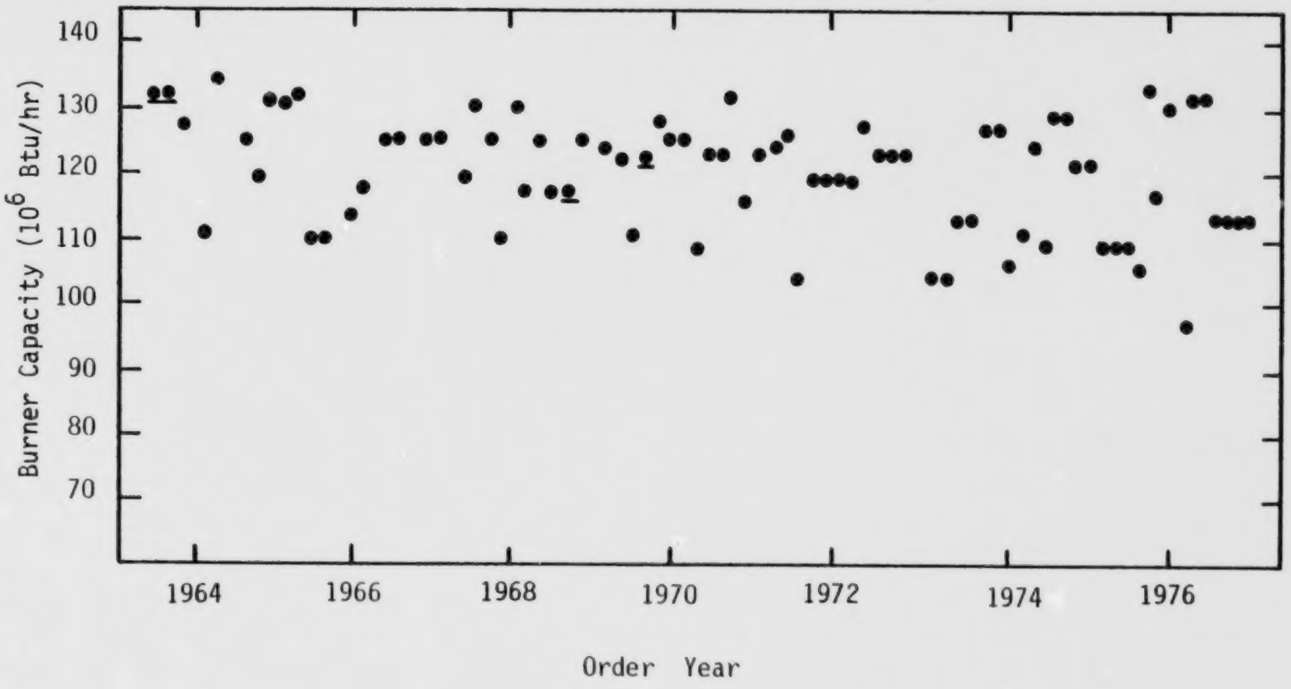


Figure 3-3. Burner Capacities for Babcock and Wilcox Pulverized Coal-Fired Boilers.

Table 3-3. Approximate Utility Boiler Demonstration Site Characteristics

Characteristic		Value
Boiler/Furnace		
Boiler Capacity:	MCR	≈350 MW
Furnace Depth		32-36 ft
Furnace Width		34-48 ft
Firing Configuration		Opposed Wall
Burner Capacity:	MCR	125 x 10^6 Btu/hr
Current Burners:	Type	Circular or Cell
Circular Burner Array	Wide x High	3 x 4 or 4 x 3
Circular Burner Spacing:	Horizontal	8-9 ft
	Vertical	8-10 ft
Fuel Supply System		
Primary Stoichiometry	at MCR	25% T.A.
Primary Temperature		150°F
Air Supply System		
Design Point Excess Air	at MCR	15%
Windbox Temperature	at MCR	550°F
Burner Pressure Drop (nominal)		4.0 in. H_2O

Table 3-4. Development DMBs Tested in Research Furnaces

FLAME TYPE	BURNER CAPACITY (10^6 Btu/hr)	BURNER ARRAY	NUMBER OF SECONDARY AIR PASSAGES	RESEARCH FURNACE
Long	10	1	1	SWS*
Long	50	1	2	LWS**
Short	12.5	1	1	SWS
Short	50	1	1	LWS
Short	100	1	1	LWS
Short	12.5	2 X 2	1	LWS

* SWS = Small Watertube Boiler Simulator Furnace: See Appendix B.

** LWS = Large Watertube Boiler Simulator Furnace: See Appendix B.

  - Burner Zone Stoichiometry (sum of primary and secondary)
  - Inner Swirl
  - Outer Swirl
  - Axial Velocity
  - Inner/Total Area
  - Inner/Total Flow Rate
- Tertiary Air Variables
  - Temperature
  - Swirl
  - Axial Velocity
  - Angle
  - Number
  - Location
  - Divergence
- Exit Geometry
  - Half Angle
  - Length/Diameter
  - Setback - Inner Secondary
  - Setback - Outer Secondary

The test results presented in this section have been organized to delineate the effects of these variables on DMB performance. However, it should be noted that many of these variables are related and cannot be varied independently. For example, the primary velocity, nozzle diameter, primary stoichiometry, coal composition and firing rate are interrelated. Variation of any one of these variables will result in commensurate variations in one or more of the others. During the experimental effort, specific design parameters were varied two ways (when possible) to more clearly determine the effects on burner performance. The tests were primarily conducted firing Utah, low sulfur high volatile bituminous coal at 100 percent of the design firing rates (full load). Additional tests were conducted with other fuels at off design point operating conditions to verify that the DMB could be operated over a wide range of conditions. These test results are discussed in Appendix C.

3.2.1 Fuel System Variables

The fuel system includes the fuel/air characteristics through the burner fuel nozzle (primary stoichiometry, etc.) and fuel nozzles geometric variables. Typical parameters include:

- Primary transport fuel/air stoichiometry;
- Nozzle size and position relative to throat;
- Flow exit velocity;
- Swirl;

Of the above variables, only the fuel nozzle size was not explicitly studied; however, over the practical range, this variable (at constant velocity) is considered to have only a secondary effect on emissions and performance.

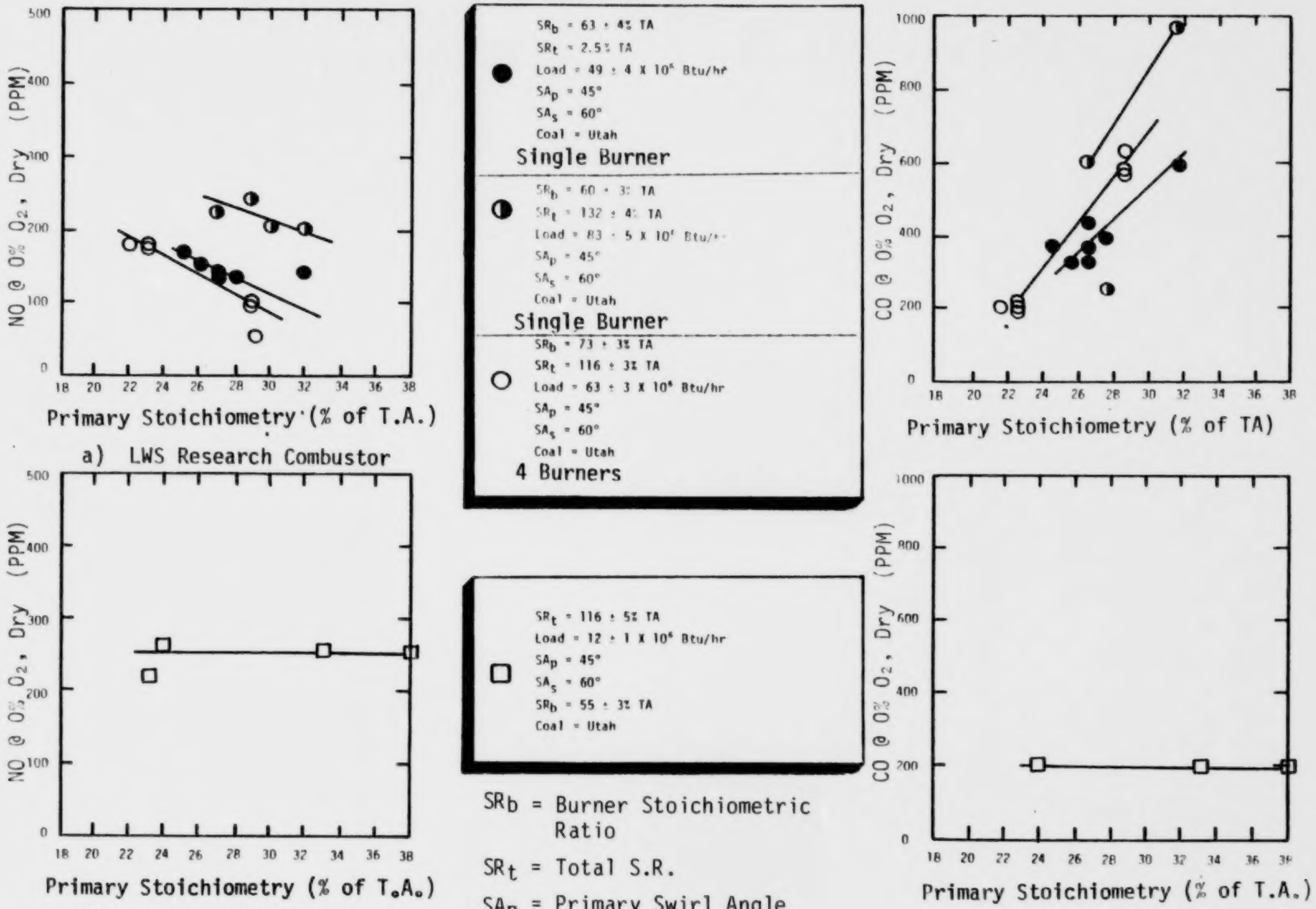
Short Flame DMB Designs

Primary Stoichiometry and Velocity

In practice, primary stoichiometry at full load is dictated by the pulverizer requirements and coal properties. Additionally, as load is decreased primary stoichiometry increases due to standard commercial pulverized operation and transport characteristics which necessitate nearly constant pulverizer airflow rates. Since commercial burners typically operate at fixed burner settings over the entire load range and with fixed geometry, then varying primary stoichiometry will also result in changes in the primary velocity. (It should be noted that if overall stoichiometry is maintained, then some reduction in secondary air velocities will also occur.) The combined effects on burner operation could result in changing the fuel/air mixing rates sufficiently that the flame characteristics could change from a short flame with ignition close to the burner exit to a long diffusion flame with ignition further away from the burner exit. These types of variations can impact NO, CO and flame stability characteristics. Since field operating experience has demonstrated that NO_x levels decrease as load is decreased, then from a design standpoint, the burner should be designed to provide (1) a stabilized flame near the exit with minimum NO_x levels at full load, and (2) flame stability over the entire load range. Therefore, the experimental effort first concentrated on establishing the impact of primary stoichiometry at full load with fixed fuel nozzle diameter designs, and then on changing burner parameters such as primary and secondary swirl angle. Load was not expressly varied since the required excess air and primary-air-to-coal ratio are strongly dependent on the site operating specifications and firing equipment.

Typical results for the three single burners and the four-burner configuration are presented in Figure 3-4. The data presented for each burner were obtained at constant load, overall stoichiometry and burner zone stoichiometry. In Figure 3-4a the results obtained in the Large Watertube Simulator (LWS) are presented, and in Figure 3-4b the results are presented which were obtained in the Small Watertube Simulator (SWS). For the burners tested in the LWS, all resulted in increasing of NO and decreasing CO as the primary stoichiometry was reduced. (As discussed later, these characteristics are completely opposite to those associated with changes in the stoichiometry at the burner throat.) Although not explicitly shown in Figure 3-4, increasing the primary stoichiometry did result in increasing the flame length; however, for these burners, over the range tested, the flame was still controlled by a weak internal recirculation zone and did not approach the long diffusion type flame. This would suggest that the effects measured on NO and CO are dominated by the changes in aerodynamics primarily altering the ignition characteristics. This hypothesis is further supported by consideration of the results obtained in the SWS where the firing wall and floor are refractory lined providing high temperature radiating sources, and due to confinement, increased high temperature recirculation near the burner and flame front. All of these factors should enhance the ignition characteristics. Note in Figure 3-4b that over the same range in primary stoichiometry as that of 3-4a and similar burner zone stoichiometries, both the NO and CO are unchanged. These results suggest that if the burner is designed to operate with the desirable flame characteristics at full load, then the design primary stoichiometry will not be an important NO_x design parameter.

Since stability is dependent on factors other than the burner alone (e.g., fuel grade, firing equipment, etc.), it is difficult in a limited effort to generate sufficient information to specify general design criteria. The results do suggest, however, that stability can be attained by proper matching



a) LWS Research Combustor

b) SWS Research Combustor

SR_b = Burner Stoichiometric Ratio

SR_t = Total S.R.

SA_p = Primary Swirl Angle

SA_s = Secondary S.A.

Figure 3-4. Effect of Primary Stocihiometry on NO/CO Characteristics for Single Secondary Burner(s).

of the primary/secondary velocities (axial and swirl) to produce the desired flame characteristics even at low burner zone stoichiometry.

Primary Air Swirl

The primary air swirl can be used to promote more rapid mixing and dispersion of the coal and volatiles within the burner firing zone. Primary air swirl should, therefore, affect both the NO_x and performance characteristics. Since rapid mixing, providing essentially plug flow at the burner throat, should result in the highest level of NO_x, the design goal should be to define the best trade-off between the mixing rate and the desirable ignition characteristics. To evaluate this parameter, experiments were conducted for several values of primary air swirl over a range in overall excess air, keeping all primary and secondary flow parameters and secondary swirl settings constant. The most extensive effort was conducted using the 50×10^6 Btu/hr single secondary burner in the LWS, wherein primary swirl was varied from 0^o (100 percent axial flow) to 60^o using an impeller type coal spreader. Typical NO and CO results are shown in Figure 3-5. (Although not shown, when the swirl angle was set at 0^o, it was not possible to obtain acceptable flame stability.) As shown in Figure 3-5, at 30^o swirl angle an acceptable flame was achieved with minimum emissions, and no measurable increase in NO_x was observed when the swirl angle was increased to 45^o. Further, increasing the swirl angle to 60^o resulted in a large increase in NO_x. Due to the limited range and scatter in the CO data, the effect of swirl angle on CO is unclear. The indicated lines represent a best judgment as to interpretation of the results. This interpretation would indicate that both the fuel/air mixing characteristics and ignition flame characteristics are changing significantly. Therefore, there should be a design trade-off between NO and CO, with lower primary swirl angle providing lower NO_x levels. Ignition stability represents the lower limit of operation, although it would not be prudent to design at that low a level. Since ignition stability plays a key role in the ability to minimize both NO and CO, the specific design point for primary swirl will depend on the site-dependent parameters.

Coal Nozzle Setback

Adjustment of the coal nozzle location with respect to the burner throat allows for rapid variations in the mixing characteristics upstream of the burner throat. As mentioned above, this can provide a trade-off between the ability

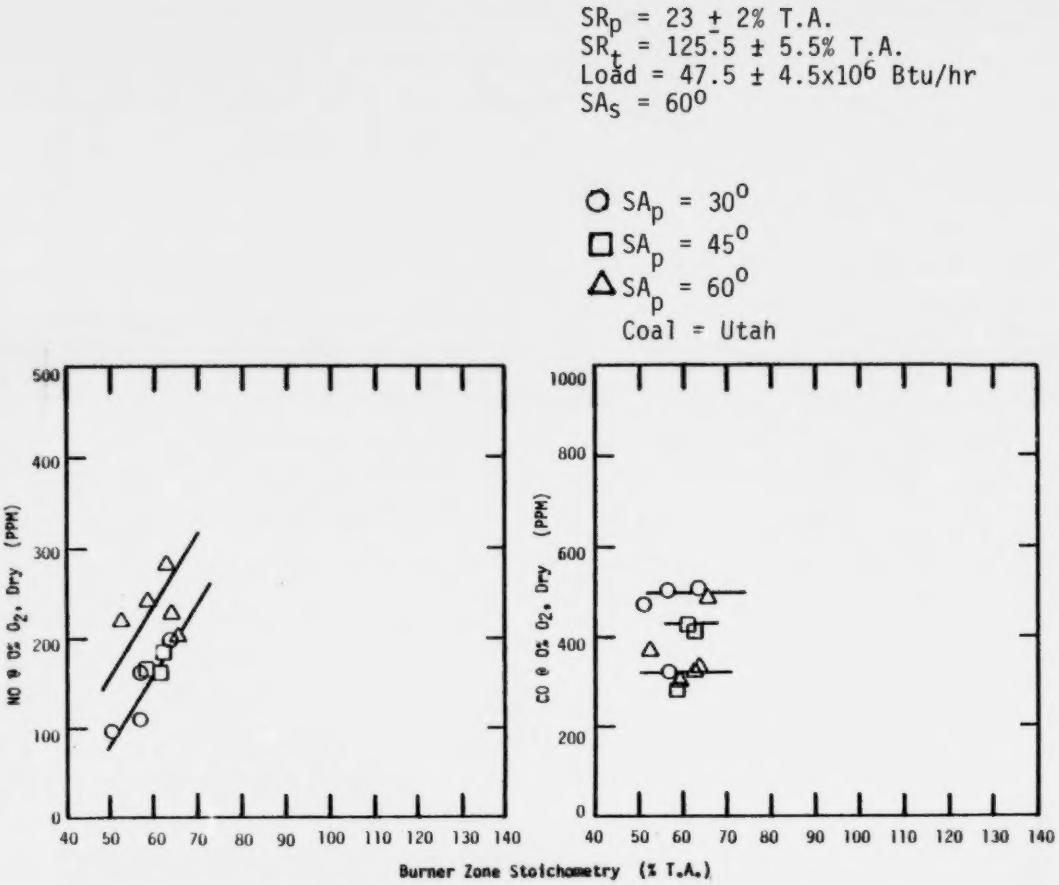


Figure 3-5. Effect of Primary Swirl Angle on NO and CO Using the 50×10^6 Btu/hr Single Secondary Burner.

to obtain improved stability characteristics and NO_x. Using the $12.5x10^6$ Btu/hr single secondary burner, the nozzle location relative to the throat was varied from zero to 3.5 inches behind the throat. The NO results presented in Figure 3-6 are indeed encouraging in that NO emissions appear to be insensitive to nozzle setback when burner zone stoichiometry is held constant. Although not explicitly shown on this figure, lower values of burner zone stoichiometry were achievable while maintaining excellent flame stability characteristics as the nozzle setback increased.

Long Flame DMB Designs

During the early phases of the program, a new commercial pulverizer was purchased and installed in the LWS system. The new pulverizer required a significant change in the air necessary for proper pulverizer operation resulting in a significant increase in the primary coal nozzle injection velocity. This change in primary velocity resulted in a large difference in the stable operating limits of the burner. An example is shown in Figure 3-7, wherein the high primary velocity the burner could be operated at a burner zone stoichiometry only as low as 90 percent of theoretical air, while for the lower velocity, stable operation could be maintained at values as low as 60 percent of theoretical air. Also of interest is that equivalent NO levels were attained at differing overall burner zone stoichiometry, supporting the contention that the slower diffusion rate results in a lower "effective" stoichiometry in the combustion region.

These results suggest that the longer the design flame, the more sensitive the burner will be to stability. The specific design value will depend on the overall boiler duty cycle requirements and pulverizer requirements since these variables set the limits of operation. The burner zone stoichiometry will be defined to meet the NO_x goals at full load operation and the primary velocity should be set for stable operation over the load range. (Lowering load results in increasing primary nozzle velocity and increasing burner zone stoichiometry.)

3.2.2 Secondary Air Variables

Short Flame DMB Designs

Secondary air was varied in order to control the burner stoichiometry and the fuel/air mixing rates at the burner throat (burner zone stoichiometry). Interacting parameters precluded the ability to study these independently, since variations in burner zone stoichiometry also result in changes in secondary air

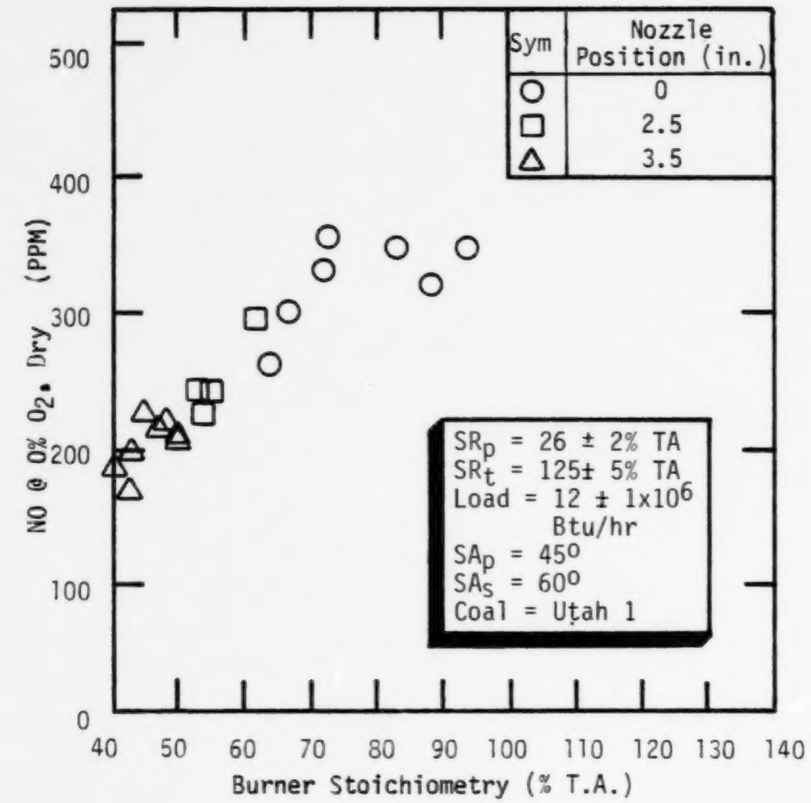


Figure 3-6. Effect of Primary Nozzle Setback on NO_x Emissions 12.5x10^6 Btu/hr Single Secondary Burners.

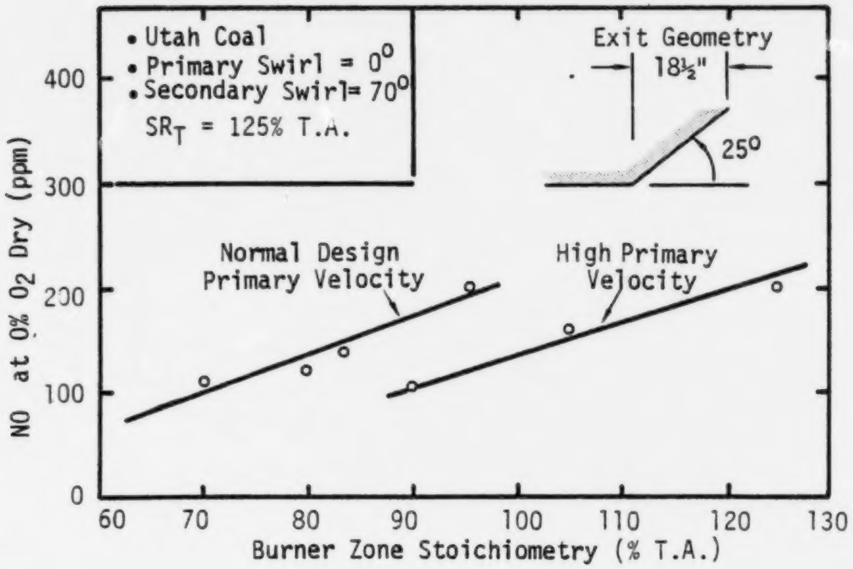


Figure 3-7. Effect of Primary Air Velocity on Operating Limits for 50×10^6 Btu/hr Dual Secondary Burner.

velocity. Further, if the overall excess air remains constant, lowering the burner air results in an increase in the tertiary air. Based on the result discussed above, however, the effect of increasing or decreasing the tertiary airflow on NO/CO was found to negligible. Therefore, for the DMB evaluated, it is expected that NO and CO variations with secondary airflow rate can be attributed to (1) the overall burner zone stoichiometry, (2) fuel/air mixing rates, and (3) flame characteristics, independent of tertiary air characteristics.

While burner zone stoichiometry was studied using all of the burners described above, the results were similar to that shown for the 50×10^6 Btu/hr single secondary burner in the upper plot of Figure 3-8. All burner results, regardless of overall excess air, show that lowering the burner zone stoichiometry results in a decrease in NO. The reduction in NO is accompanied by no increase in CO until the primary/secondary flows can no longer provide adequate mixing and flame characteristics, wherein the CO then rapidly increases. While the flame is still stable initially, continued reduction in burner air eventually leads to instability and flameout. In comparing the CO and NO characteristics, note that the impact of burner zone stoichiometry on NO remains unchanged even as the CO markedly increases. This suggests that the NO formation is being controlled principally by the burner zone stoichiometry and not by the primary/secondary flow mixing rates. The conditions producing the increase in CO shown in Figure 3-8 were found to be strongly dependent on the burner settings, as well as design velocity. In fact, at full load it was possible to obtain minimum CO and NO over a wide range in excess air levels by simply varying both burner zone stoichiometry and swirl conditions. However, in a field installation, it may not be feasible to adjust these parameters as boiler and firing conditions change. An example of varying the secondary swirl is shown in Figure 3-9. Although not shown on the figure, at zero swirl the flame could not be stabilized, even at high burner zone stoichiometry, and resulted in flame-out. As secondary swirl was increased to 30°, it was possible to achieve a stable flame over burner zone stoichiometry of about 70 to essentially unstaged conditions of 120 percent of theoretical air. Lower burner zone stoichiometries were possible as the secondary swirl angle was increased. Note that except for 30° the NO characteristics were essentially unchanged with secondary swirl angle, while the burner efficiency (CO) was improved by increasing secondary swirl.

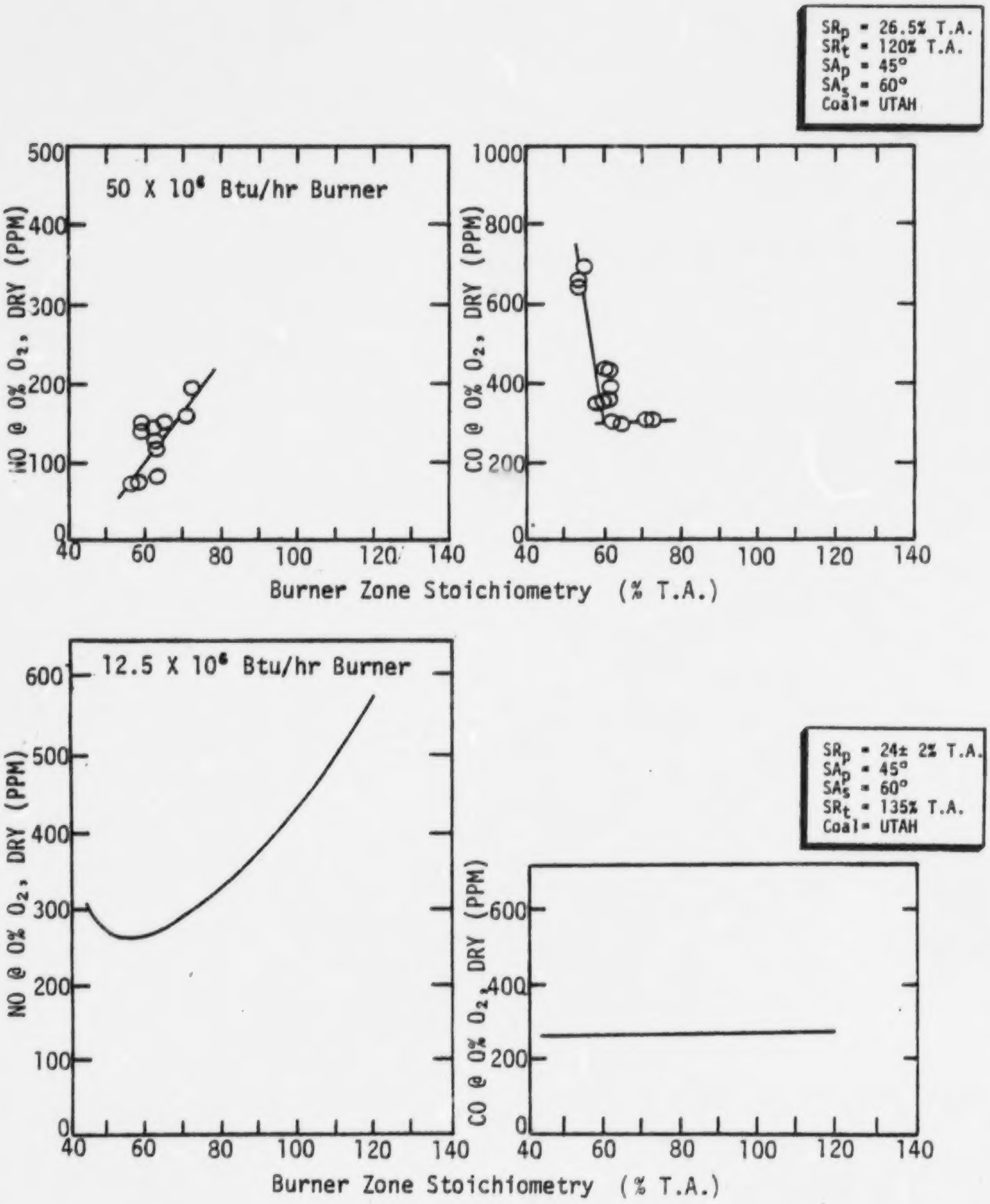


Figure 3-8. Effect of Burner Zone Stoichiometry on NO and CO Emissions with Two Single Secondary - Short Flame Burners.

$SR_p = 26.5 \pm 2.5\%$ T.A.
$SR_t = 120 \pm 10\%$ T.A.
Load - $48 \pm 3\text{x}10^6$ Btu/hr
$SA_p = 45^0$

○ $SA_s = 30^0$
□ $SA_s = 45^0$
△ $SA_s = 60^0$

Coal = Utah

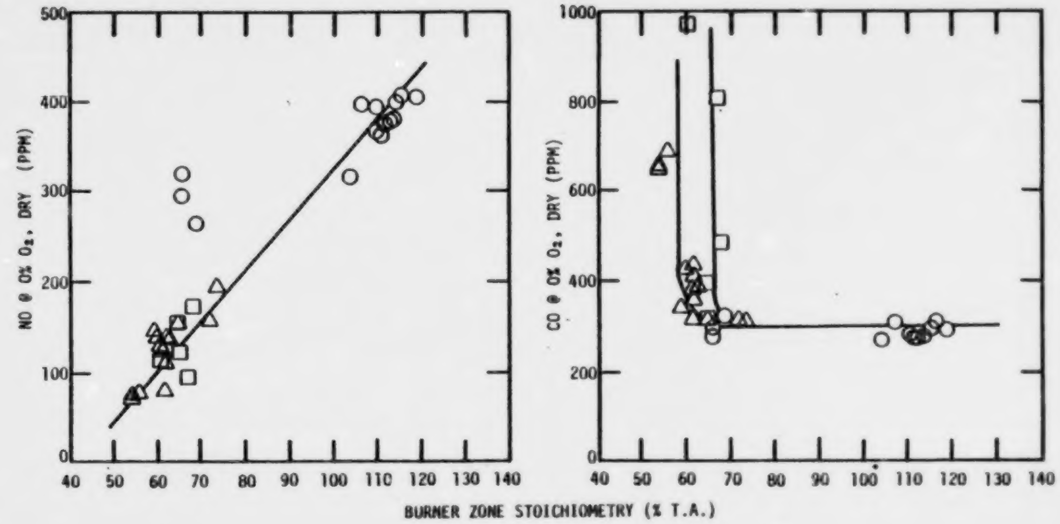


Figure 3-9. Effect of Secondary Swirl Angle on NO and CO Characteristics ($50\text{x}10^6$ Btu/hr Single Secondary Burner).

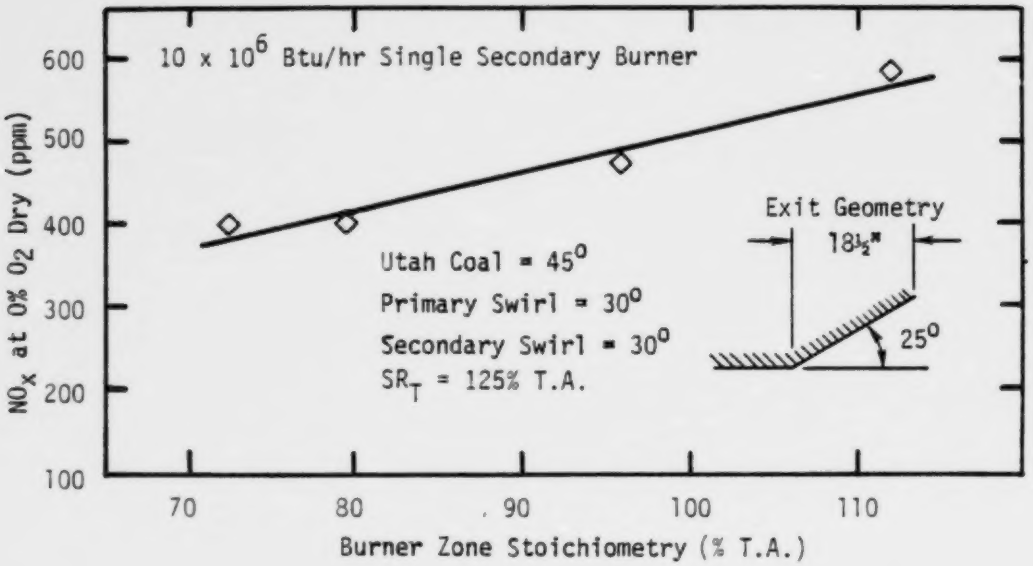
The results shown for the 12.5×10^6 Btu/hr burner (the lower plot of Figure 3.8) support the thesis that for the short flame burners, the burner zone acts as a well mixed first stage system. These results represent a cross plot of data obtained at various burner zone stoichiometries over a range of excess air levels. As shown, the NO decreases as burner zone stoichiometry is decreased until a minimum is reached and then with further decreases in stoichiometry, NO increases. These results are very similar to those found in small-scale testing with plug flow reactors wherein the staging was well controlled. It is felt that for the larger burners, a minimum would also occur if burner zone were lowered beyond that evaluated (≈40% T.A.). However, this would be very difficult since at 40% T.A., the air flow rate through the secondary air passages is only 1/5 of the unstaged value.

Long Flame DMB Designs

The burner zone stoichiometry was varied identically to that described for the previous burners. Typical results for the two scales of burners are provided in Figure 3-10. The sensitivity of NO to the burner zone stoichiometry is approximately the same as that of the shorter flame burners discussed previously. Although for the same burner capacity, the longer flame will achieve NO levels equal to the short flame design only at higher overall burner zone stoichiometry. This could be the result of higher mean temperatures with the short flame or higher O_2 concentrations during devolatilization. The specific burner zone stoichiometry design point will depend on the boiler and coal characteristics, as well as the desirability of providing a long or a short flame.

3.2.3 Tertiary Air Variables

The DMB utilizes both secondary air passages and tertiary air ports. Physical separation of the tertiary air from the burner air ensures that the burner zone will operate overall at substoichiometric conditions and final staging is then controlled by interaction with the outboard tertiary air. The ability to evaluate the effect of tertiary air independent of the secondary air is not possible with fixed burner geometry. That is, if the overall excess air remains constant, then changing the secondary airflow rate results in variations in (1) burner zone stoichiometry, (2) secondary air velocities, and (3) tertiary air velocity. Consequently, changing second air can affect the mean stoichiometry in the burner throat, burner fuel/air mixing rates, flame characteristics and staged air mixing characteristics. While physical separation of the para-



Primary Air at Normal Design Velocity (26 ± 2%)

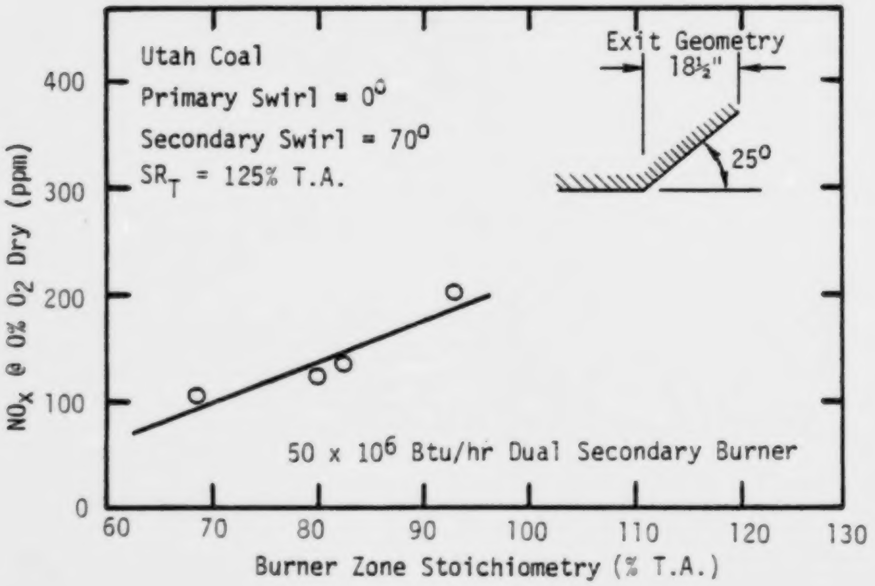
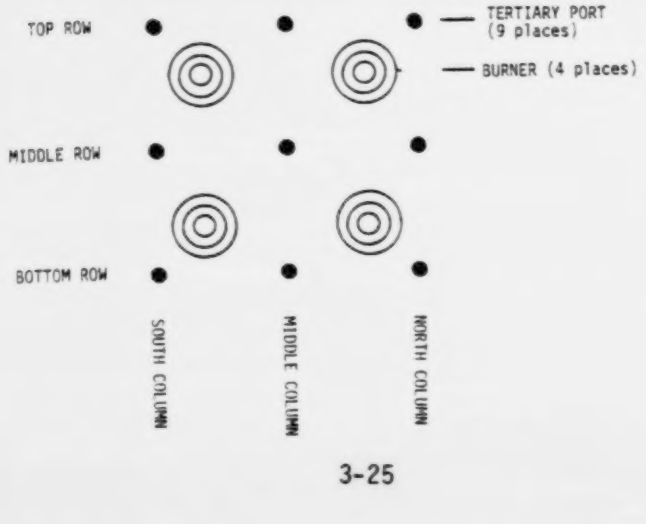


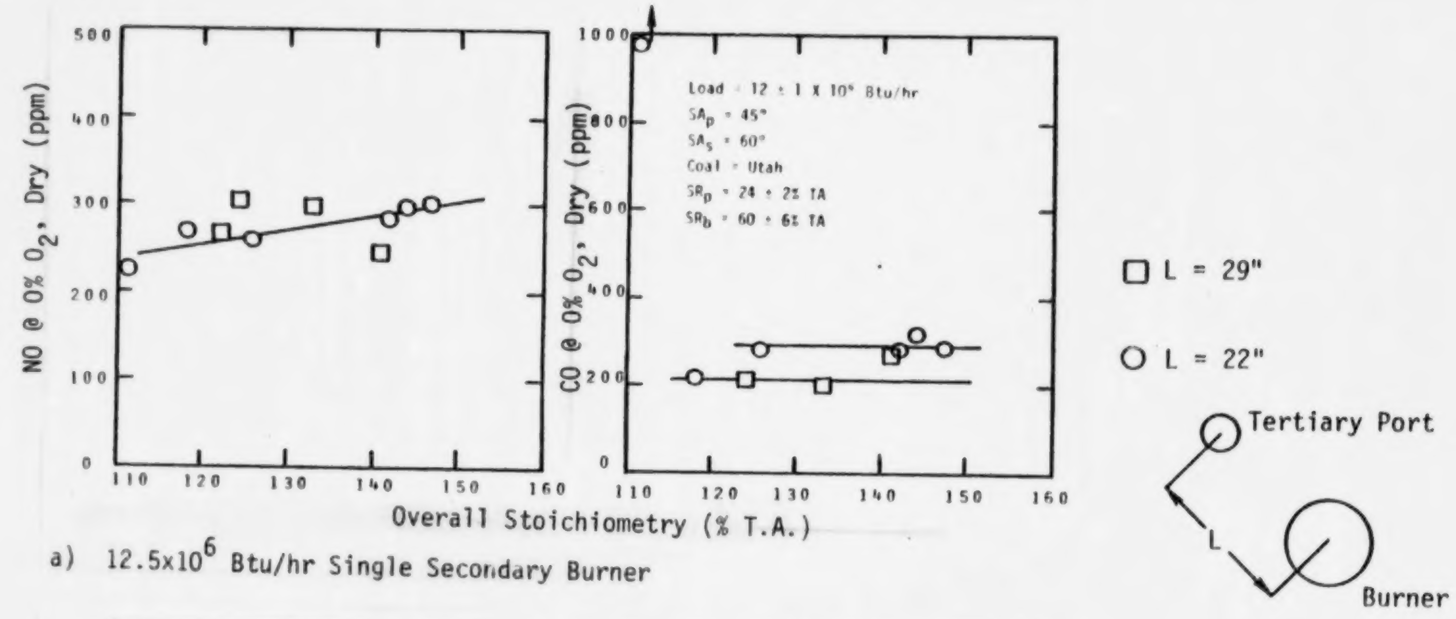
Figure 3-10. Effect of Burner Zone Stoichiometry on NO Characteristics for Long Flame Burners.

meters is not completely possible, sufficient parameters, both flow and geometric, were varied to heuristically deduce gross mechanisms. The tertiary air parameters were only varied using the short flame DMB designs.

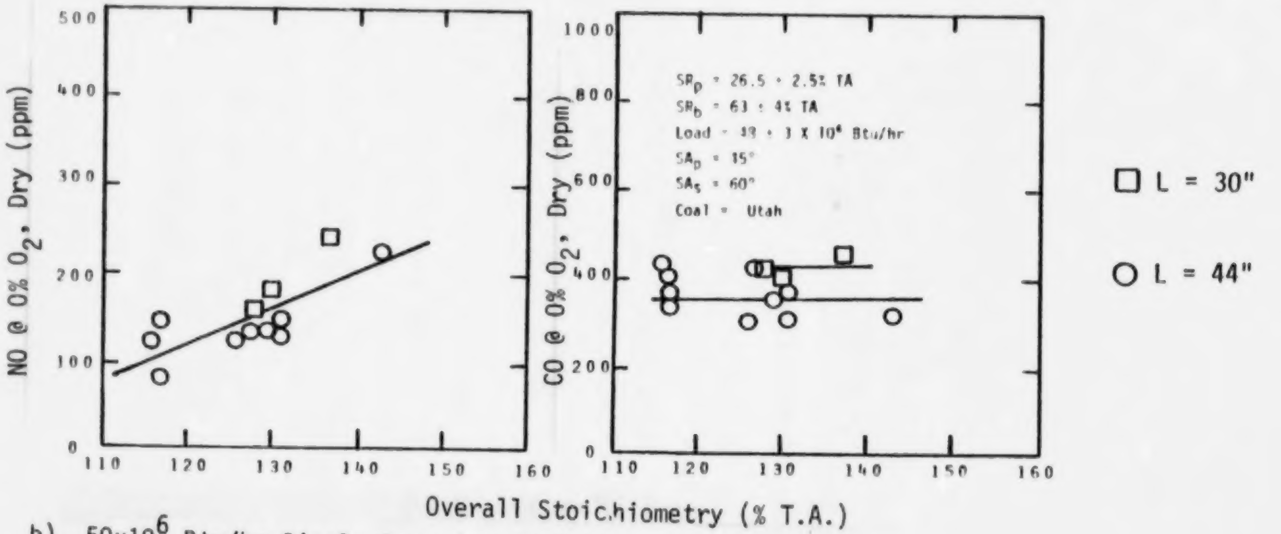
Variations in tertiary air port location and injection velocity were evaluated with both the single and four-burner configurations and jet velocity was evaluated in two single burners. The impact on NO and CO characteristics was studied at two tertiary port locations using the 12.5 and 50×10^6 Btu/hr burners. The results shown in Figure 3-11 show that for both burner sizes the port locations affected CO levels, while NO appeared unaffected. Of particular interest is that changing the overall stoichiometry (as well as tertiary air injection velocity) at a fixed tertiary port location has little effect on the CO characteristics. This suggests that the overall mixing between the burner fuel/air products is insensitive to the tertiary air velocity while physically moving the ports closer affects only the overall mixing. Since the injection velocity decreases rapidly with distance from the nozzle, it is possible that the changes in effective shear mixing between the burner zone products and the tertiary jets is not significant over the practical range evaluated. The above results relative to the impact of tertiary air velocity on mixing characteristics are further supported in Figure 3-12, wherein tertiary jet velocity was varied independently using the 12.5×10^6 Btu/hr single and multiburners. Note that in both cases, variation in velocity by as much as a factor of two did not affect NO characteristics in either burner. A complete set of data were available from the four-burner tests, and they show that CO was also constant.

The effect of tertiary air port location was also studied in the four-burner multiburner configuration shown in the sketch below. Tests were



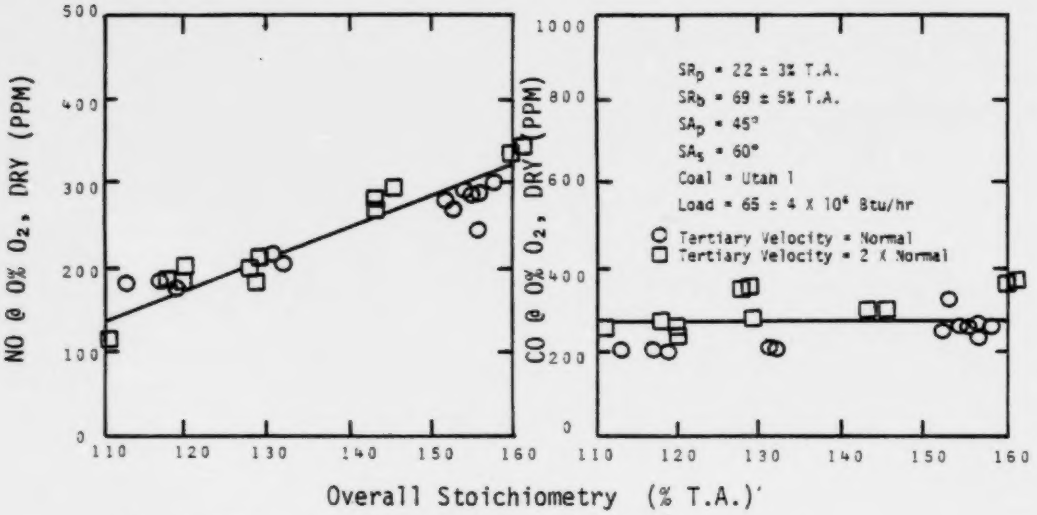


a) 12.5×10^6 Btu/hr Single Secondary Burner

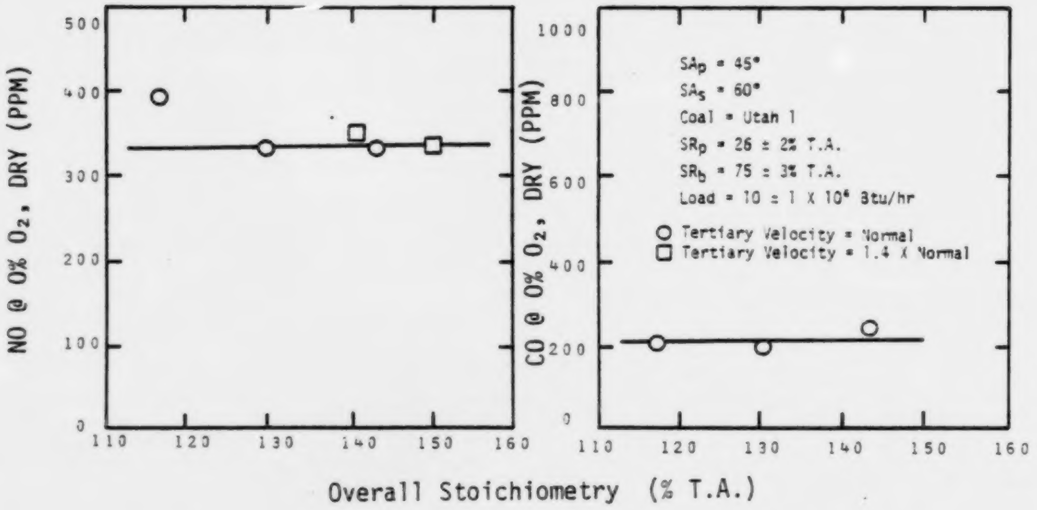


b) 50×10^6 Btu/hr Single Secondary Burner

Figure 3-11. Effect of Tertiary Port Location on NO & CO Characteristics.



a) Four Single Secondary 12.5 X 10^6 Btu/hr Burners



b) Single Burner - 12.5 X 10^6 Btu/hr Single Secondary Burner

Figure 3-12. Effect of Tertiary Air Velocity on NO and CO Characteristics.

conducted with various rows and/or columns out-of-service to define the sensitivity of NO and CO, or number and location of the tertiary ports. The results presented in Figure 3-13 again show that within experimental accuracy, the NO characteristics were completely unaffected by the variations in tertiary air velocity or tertiary port location. The general conclusion from this is that for these short flames, the burner zone is essentially premixed and burns at the burner zone stoichiometry, and, therefore, NO is formed in the burner flame zone before substantial mixing with the tertiary air. While NO was not affected by tertiary port location in the multi-burner configuration, the variations in CO were dramatic. Note that any change in tertiary ports out-of-service were accompanied by an increase in CO. This result reveals that the tertiary port location relative to the burners has a strong impact on the overall mixing in the lower furnace, and that unsymmetric port location will result in incomplete CO burnout in the high temperature region of the furnace.

3.2.4 Burner Exit Geometry

All of the development DMB designs utilized cast refractory exits. The effects of burner exit geometry on NO emissions were evaluated on the long flame DMB designs.

Two differing burner exit scaling methods were evaluated: (1) direct geometric scaling wherein the quarl length was extended, keeping the divergence angle constant; and (2) a combination of increasing the divergence angle and shortening the quarl length. In the geometric scaling case, extending the quarl should cause the flow to expand to more nearly fill the exit, resulting in more rapid mixing of burner fuel/air and shortening of the flame length. The results, obtained using the 10×10^6 Btu/hr burner presented in the upper plot of Figure 3-14, show that extending the quarl significantly increased the NO levels suggesting more rapid fuel/air mixing. In addition, the longer quarl length resulted in significantly improved stability characteristics. The quarl geometry which resulted in the best stability characteristics was also evaluated at the 50×10^6 Btu/hr burner scale. The large burner was stable over the operating range shown in the lower plot of Figure 3-14, even at the high primary air velocities. Since typical commercial systems utilize quarl lengths considerably shorter than 18.5 inches, an attempt was made to try to obtain equivalent flame characteristics by both shortening the quarl length and increasing the quarl angle. With this configuration, although the NO levels were simlar to the

SSB - 12.5 x 4 Closed Tertiary Analysis

- ○ BASELINE (all ports open)
- □ BOTTOM ROW CLOSED
- △ MIDDLE ROW CLOSED
- ◇ MIDDLE ROW & COLUMN CLOSED
- ⬡ SOUTH COLUMN CLOSED
- ▽ NORTH COLUMN CLOSED
- ◊ MIDDLE COLUMN CLOSED

SR_p = 21 ± 2% T.A.
SR_b = 62 ± 5% T.A.
Load = 67 ± 5×10^6 Btu/hr
SA_p = 45°
SA_s = 60°
Coal = Utah 1

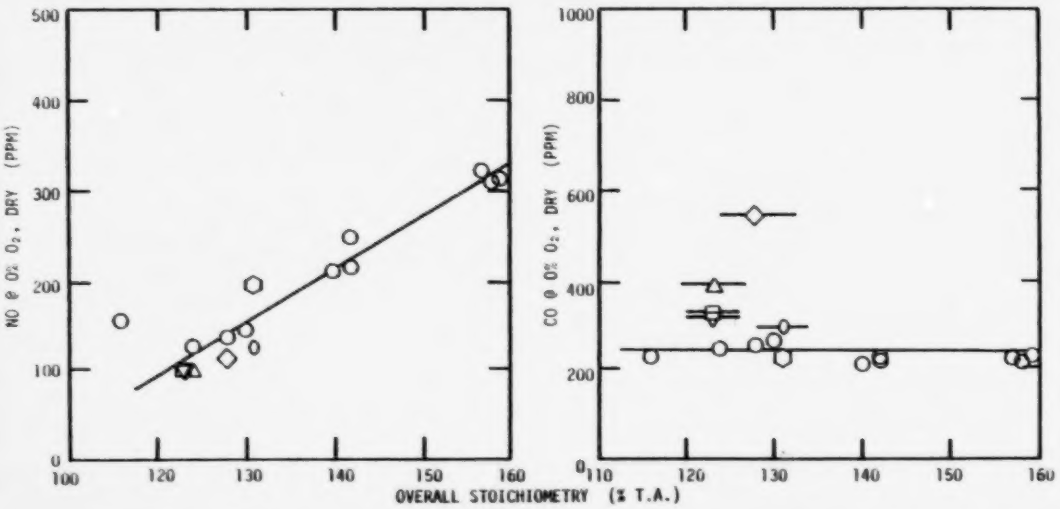
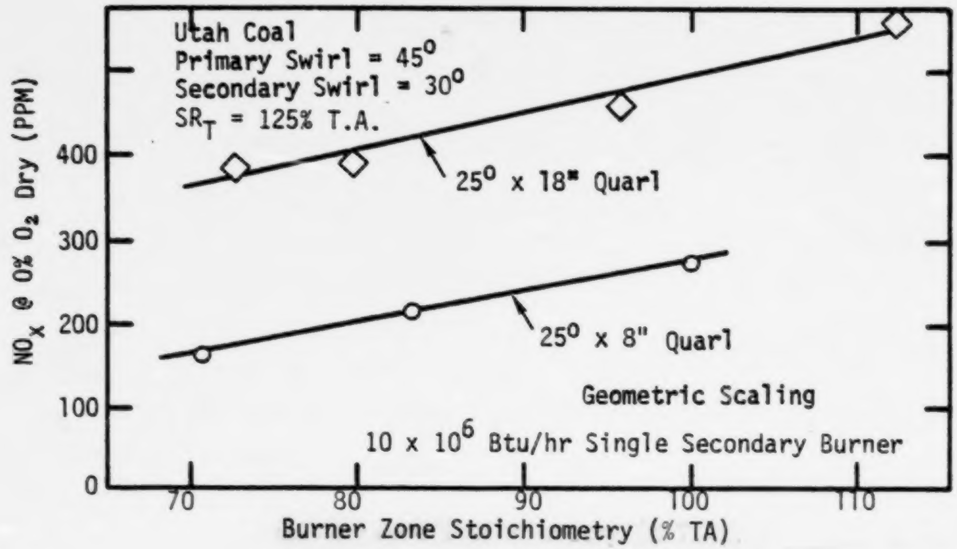


Figure 3-13. Effect of Tertiary Port Location and Number on NO/CO Characteristics in a Multiburner Configuration.



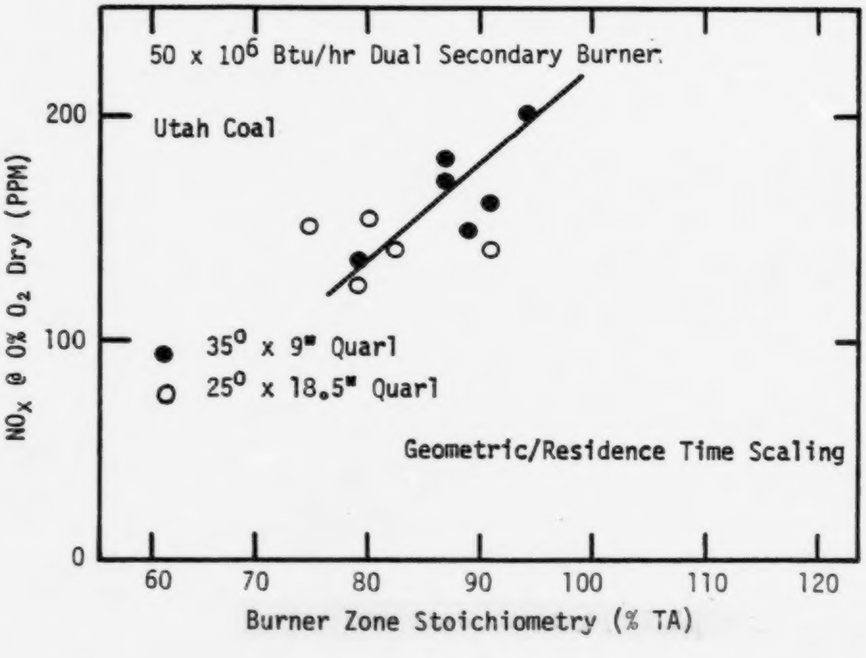


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Figure 3-14. Effect of Quarl Geometry on NO Characteristics.

25^{o} x 18.5 inch quarl as shown in Figure 3-14, the stability characteristics were not acceptable.

These results suggest that for stability control, a long flame burner design operating at low burner zone stoichiometries may require a longer quarl length than is normally utilized in commercial practice. This requirement could significantly limit retrofit of long flame systems in existing boilers.

3.2.5 General DMB Design Requirements

The data presented in the previous sections have shown that distributed mixing burners can be designed and operated to produce a wide range of flame shapes and emission levels. This section summarizes these data and identifies the general requirements for distributed mixing burners designed for field boilers.

The fuel system design has significant influence on flame shape and length. Nozzles in which the coal and primary air are injected axially tend to produce slow fuel/air mixing and long diffusion flames. Nozzles with impellers cause more rapid fuel/air mixing which tends to produce shorter more compact flames. Long diffusion flames can be used in boilers with large furnace depths such as opposed wall-fired units. The NO emissions from this flame type were generally lower than those for short flames. However, this may be valid only for the specific burner configurations tested and is not necessarily a general result. Smaller furnaces such as those in single wall-fired units require shorter flames to avoid flame impingement on the rear wall. The test results have shown that the short flame distributed mixing burner can be optimized for low NO emissions, but that deeper staging (low burner zone stoichiometry) is required. A distributed mixing burner could be designed to vary flame shape by incorporating some mechanism for changing the spreading characteristics of the primary stream into the secondary airstream. For burners designed with annular tangential inlets, this parameter is most easily varied by burner setback which increases the residence time in the primary/secondary mixing zone upstream of the burner throat. For burners designed with a central fuel nozzle, impeller blades can be used to increase the coal spreading rate. In the short flame distributed mixing burners, optimum results were obtained with 45^{o} impeller blades. However, the tests were of limited duration and did not evaluate erosion which may limit impeller life in burners operated for extended periods as in field boilers. A significant amount of data are not available to evaluate the effects of primary velocity. Consequently, distributed mixing burners should be designed to allow primary velocity to be varied during burner tuning. This could be achieved by designing for a low primary velocity and providing sleeves or other devices to reduce the nozzle exit diameter. Alternatively, a conical nozzle exit such as the Foster Wheeler variable velocity nozzle could be used (see Appendix D).

The air system design has significant influence on NO and CO emissions. Dual secondary air channels provide additional control over fuel/air mixing rates and are thus preferred over single secondary burner design. Burner zone stoichiometry is the major parameter controlling NO for the DMB and the lower limit will be dictated by CO or flame stability. Flame stability can be enhanced at lower burner zone stoichiometries, however, by increasing the secondary swirl without impact on NO. Since the optimum value of swirl and burner zone stoichiometry will be dependent upon the burner characteristics, swirl device, and coal composition, as well as the overall duty cycle, DMBs should be designed with sufficient flexibility to vary these parameters. Although not explicitly discussed, the flame stability characteristics are also influenced by the primary air velocity. Capability to vary the primary air velocity will also provide an added means for operating at low burner zone stoichiometry while maintaining low CO levels. Tertiary jet velocity does not appear to be an important design parameter for NO or CO, although port location has a significant impact on CO burnout. For DMBs designed for field boilers, the location of tertiary ports will be determined in part by the burner-to-burner spacing and other furnace and windbox design parameters.

The burner exit configuration plays a key role in flame stability. For most of the distributed mixing burners tests, the primary and inner secondary setbacks were zero, the quarl divergence half angle was 25 degrees and its length was equal to the throat diameter. If this exit configuration is applied to DMBs for field boilers, the resulting quarl length and overall diameter will be considerably larger than current commercial practice. Consequently, a shorter quarl length will probably be required. The test results discussed above showed that flame stability decreased with decreasing quarl length, and it may be necessary to increase the coal nozzle setback or swirl on the primary or secondary air streams to achieve satisfactory flame stability in a burner with a short quarl.

These general burner design requirements are summarized in Table 3-5. Several of the parameters are listed as site-dependent (S.D.) and must be based on the characteristics of the field boilers. It is possible that the nominal design point values may need to be varied for optimum burner performance. Thus, DMBs must be designed to be "flexible" so that the design parameters can be adjusted as required for optimum performance. Table 3-5 also lists the range of variables which should be examined as part of burner performance optimization tests.

3.3 Commercial Burner Practice

The commercial burner manufacturers have developed their products to be compatible with current boiler designs and to meet specific performance, emission and safety standards. The prototype DMBs must meet these same standards. This section discusses the state of the art commercial burner design practice for standard (Pre-NSPS) burner as well as low NO_x (Post-NSPS) burners.

Standard burners are normally designed to produce a compact intense flame compatible with small furnaces. They utilize a central or annular coal nozzle through which pulverized coal is pneumatically conveyed into the furnace. The nozzles are designed to rapidly mix the coal with the combustion air which is introduced with swirl through an annulus surrounding the coal nozzle. The result is an intense turbulent flame. This burner design, which operates with all the air introduced through the burner, produces a highly desirable short stable flame, but also produces NO_x emissions in excess of current New Source Performance Standards (NSPS). Standard burners are discussed in the following subsection.

To meet NSPS, several burner manufacturers have recently developed new burners. These "low NO_x burners" vary in design details, but generally provide two or more concentric combustion air annular passages to distribute fuel/air mixing over a larger portion of the furnace volume. Flames from these burners tend to be less intense, some are longer and all have NO_x emissions on the order of half of the pre-NSPS burner emissions. Low NO_x burners are discussed in Section 3.3.2.

3.3.1 Standard Burners

The major functions of the burners are to provide stable and rapid ignition of the fuel/air mixture, and to promote combustion in such a way as to complete

Table 3-5. Prototype Burner Design Parameters.

Parameter	Nominal Design Point	Testing Range
Fuel System		
Primary Temperature (°F)	S. D.*	130-180
Primary Stoichiometry (% T.A.)	S. D.	17-30
Primary Velocity (ft/sec)	75	50-90
Primary Swirl	45° Vanes	Variable
Air System		
Secondary		
Temperature (°F)	S. D.	400-650
Burner Zone Stoichiometry (% T.A.)	50-70	40-120
Inner Swirl	Variable	Variable
Outer Swirl	Variable	Variable
Axial Velocity (ft/sec)	60	50-90
Inner/Total Area	0.33	Variable
Inner/Total Flow Rate	0.33	Variable
Tertiary		
Temperature (°F)	S. D.	400-650
Swirl	None	—
Axial Velocity (ft/sec)	50	40-60
Angle (degrees)	0	—
Number	4	—
Location (radius/throat diameter)	2.2	S. D.
Divergence (degrees)	0	—
Exit		
Half Angle	25	Variable
Length/Diameter	1.0	S. D.
Setback - Inner Secondary (inches)	0	Variable
Setback - Outer Secondary (inches)	0	Variable
Operational Variables		
Capacity (10^6 Btu/hr)	S. D.	—
Turndown (% capacity)	S. D.	≈50
Overall Stoichiometry (% T.A.)	S. D.	100-150

*S.D. = Site Dependent

combustion in a minimum furnace volume without impingement of the flame upon the furnace walls. In some large boilers which do not use circular burners, the burners have the additional task of regulating the steam superheat temperature or the temperature of the reheated steam, since adjustment of the burners may be used to control heat transfer to the furnace tube-walls. The amount of air in the primary coal/airstream varies with the heating value of the coal, its moisture content, and the specific pulverizer characteristics and may represent 15-20 percent of the total airflow. The primary air temperature is normally around 160°F and the velocity is usually about 60-100 ft/sec. The secondary airflow, which is supplied through the annulus surrounding the fuel pipe, has a nominal velocity of approximately 100-130 ft/sec., and a temperature of 600°F. The rate of fuel/air mixing, and therefore, the combustion intensity and ignition characteristics, are normally controlled by the method of fuel injection and by the introduction of a tangential component to the secondary airflow. Wall-fired boilers which predate NO_x considerations were normally fitted with burners which produced rapid fuel/air mixing and high intensity short flames. Without exception, the primary function of current burner designs is to establish ignition near the burner throat and to maximize combustion efficiency within the burner zone. This requires judicious design of the burner fuel and air systems to ensure that the fuel/air mixing and ignition requirements are met. Burner design has evolved over time, as have pulverized coal-fired steam generator requirements, with each manufacturer developing its own specific design criteria. A large body of operating data has been obtained covering a wide range of conditions over the course of many years and the high operating efficiencies and reliability of modern boilers attest to the success of this approach.

Coal supply characteristics for pulverized coal-fired burners are listed in Table 3-6. Coal must be pulverized to the point where particles are small enough (surface per unit of mass sufficient large), to assure proper combustion. Primary airflow must be sufficient to avoid settling in the feed system, but is also determined by the coal quality and pulverizer requirements. Excessive velocities must be avoided to minimize conduit erosion, high pressure drop and the corresponding high fan power requirements. Most large coal transport systems are sized for velocities in the transport line and burner nozzle of 75-90 ft/sec. at maximum pulverizer capacity, with weight ratios of

Table 3-6. Coal Supply Characteristics for Pulverized Coal-Fired Burners

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Pulverized Coal Fineness

Fixed Carbon (%)	86-98	78-86	69-78	<69	<69	<69
Heating Value (10^3 Btu/lb)	—	—	—	>13	11-13	<11
% Through 200 Mesh	80	75	70	70	65	60

Safe Primary Coal/Air Temperature*

	Exit Temperature	
Fuel	K	F
Lignite	322-333	120-140
High Volatile Bituminous	338	150
Low Volatile Bituminous	338-352	150-175
Anthracite	366	200
Petroleum Coke	366-394	200-250

*To avoid premature ignition

primary-air-to-coal of from 1.2:1 to 2:1. At reduced ratings, the need to maintain pipe velocities or pulverizer airflow prohibits reduction of transport airflow in direct proportion to the reduction of the coal feed rate. For example, at 40 percent of mill capacity, the primary-air-to-coal ratio may be increased from 2:1 to 3.5:1 to maintain a minimum velocity of 50 ft/sec. The primary airstream typically accounts for about 20 percent of the total combustion air, but percentages may be as low as 10 or as high as 40 depending on the burner design and load in relation to full capacity.

The pulverizer air is generally heated to evaporate surface moisture from the coal and control the coal/air mixture temperature. Proper pulverizer operation requires the coal to be dry and powdery. The pulverizer air temperature required to achieve this condition primarily depends upon the amount of moisture present. Final temperature of the coal/air mixture is determined by a balance between burner ignition requirements and preignition problems. Table 3-6 also lists safe temperatures for coal/air mixtures from pulverizers which generally prevent preignition. To assure stable ignition, the temperature of the primary air and coal leaving the pulverizer must be at least 130°F for units burning coal with 30+ percent volatile matter, going up to temperatures as high as 180°F where the volatile matter of the coal is as low as 22 percent.

A wide variety of coal nozzles are used by the various manufacturers. They are generally designed to provide rapid mixing of the coal/primary air with the surrounding secondary air. Typical nozzle configurations are illustrated in Figures 3-15 and 3-16. This secondary air is normally preheated to 500-600°F and the axial velocity is approximately 100 ft/sec. The secondary air is introduced with a swirl component to stabilize the flame. Flow from the windbox is controlled by the register which varies the airflow rates and generates swirl as shown in Figure 3-17. Flow rate and swirl are generally not independent on most burners, although some have separate dampers.

The level of excess air required in pulverized coal-fired boilers depends upon the following factors:

- Burner Performance - Most burners require at least 15 percent excess air to assure complete combustion.
- Slagging and Fouling - The ash fusion temperature is usually lower under reducing conditions. The excess air level must be sufficient

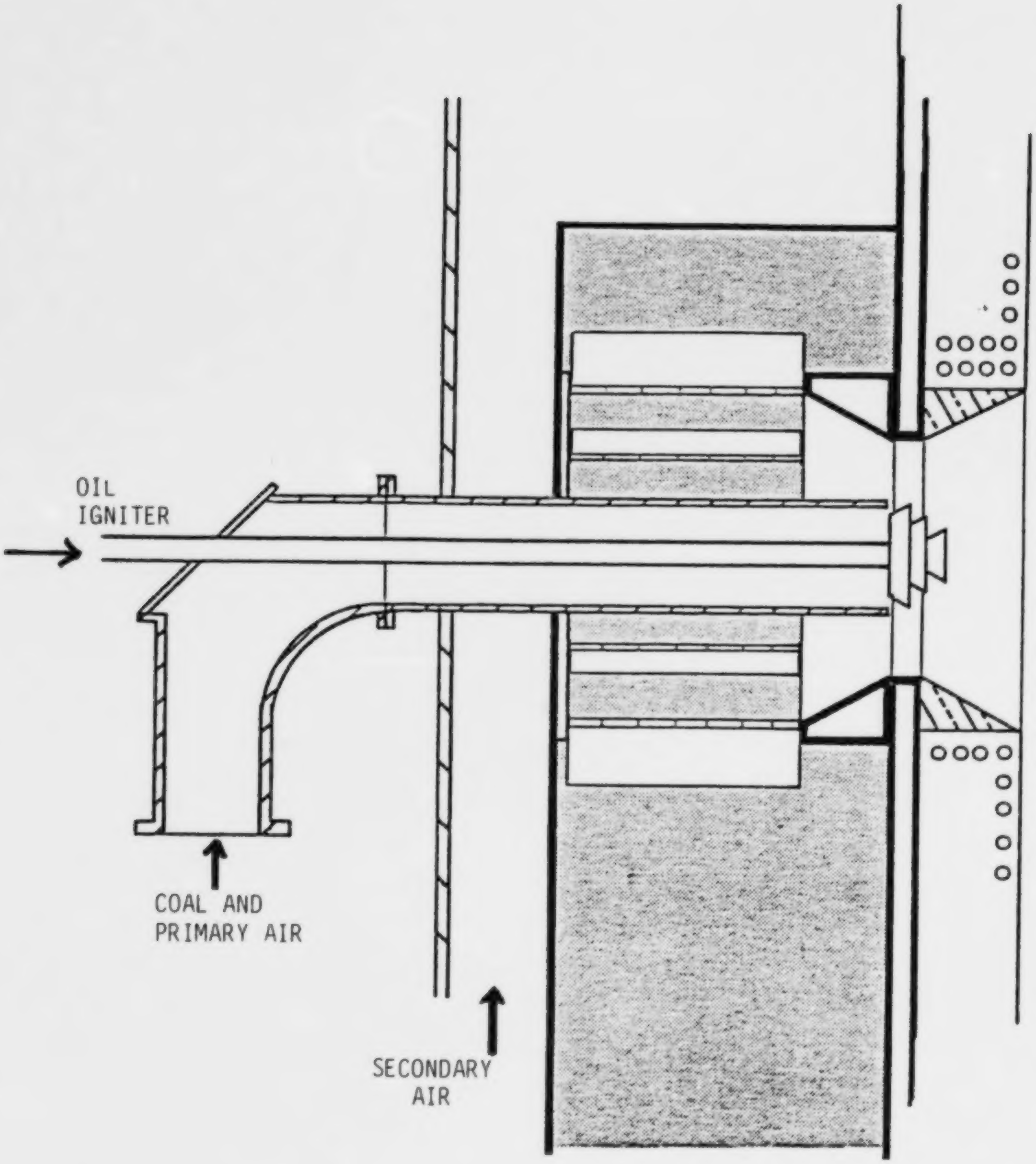

OIL
IGNITER
COAL AND
PRIMARY AIR
SECONDARY
AIR

Figure 3-15. Typical PC Burner Design.

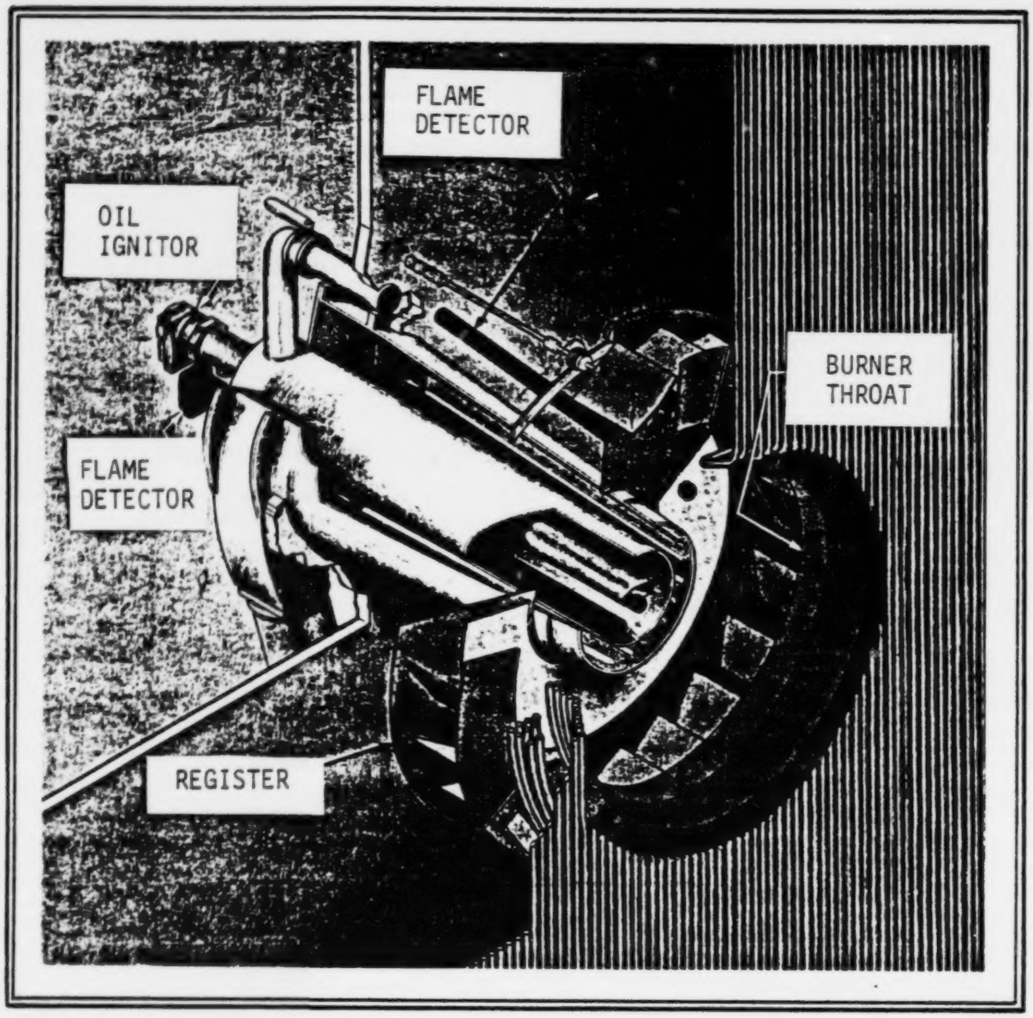

FLAME DETECTOR
OIL IGNITOR
BURNER THROAT
FLAME DETECTOR
REGISTER

Figure 3-16. Foster Wheeler Intervane Burner.

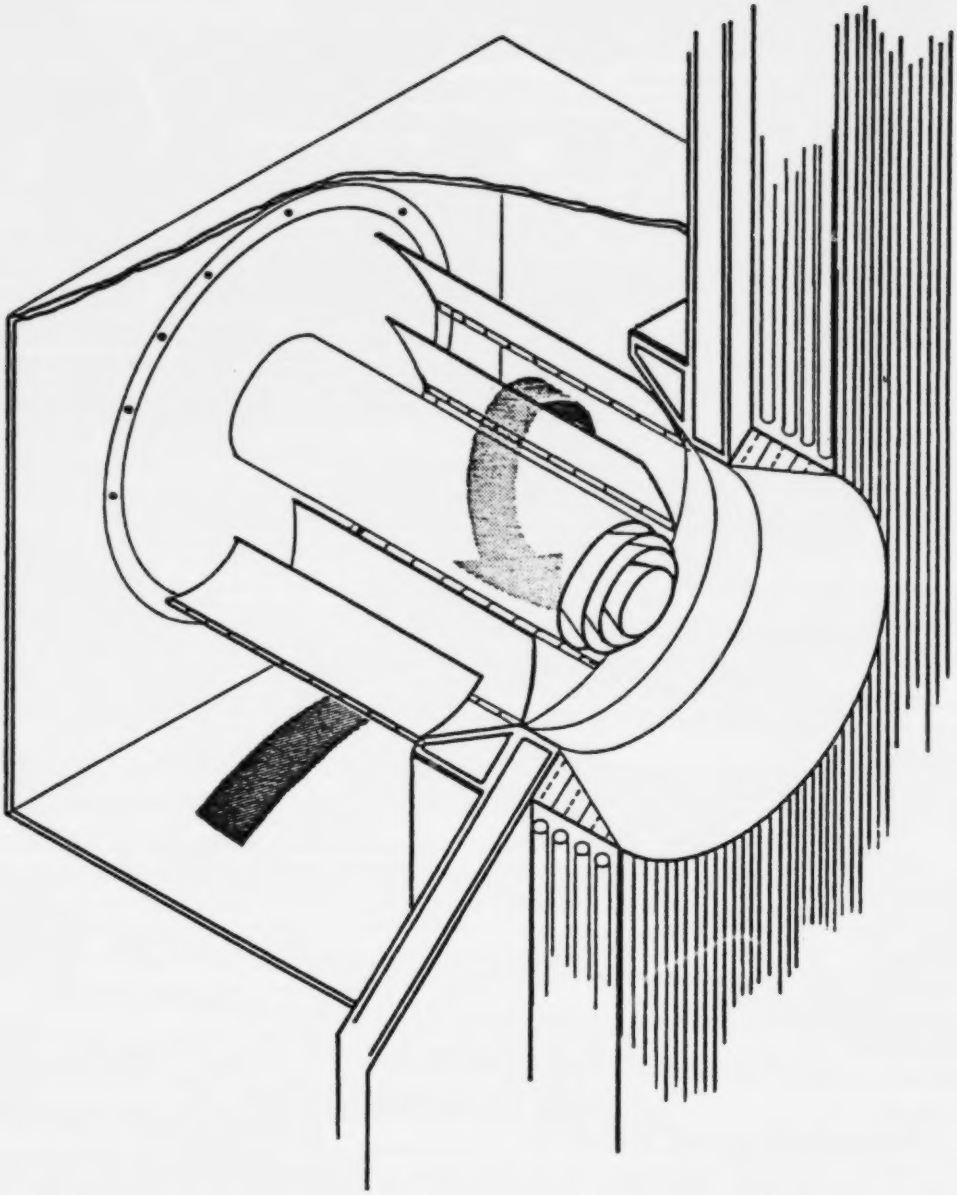


Figure 3-17. Typical Air Registers.

to ensure oxidizing conditions on the furnace walls and throughout the convective section.

- Steam Temperature - The steam temperature produced by the boiler depends upon the gas flow rates through the boiler.
- Air Distribution - Maldistribution of air among the burners can cause some burners to operate with lower excess air than others. The overall excess air must then be increased to ensure complete combustion from each burner.

Low excess air operation is generally desirable because it improves boiler efficiency (by reducing flue gas enthalpy losses) and generally decreases NO_x emissions. Excess air can be reduced by improving the air distribution among the burners. One approach used by Babcock and Wilcox is to horizontally divide the windbox into several separate plenums feeding separate burner rows. Flowmeters and dampers in each plenum allow the air distribution to be adjusted to provide approximately equal stoichiometries to all burner rows. Another approach currently being used by Foster Wheeler involves a noncompartmented windbox and flowmeters on each burner. Each air register is surrounded by a perforated plate and a sliding sleeve which varies the effective open area of the perforated plate. The pressure drop across the perforated plate is an index of flow rate, and the sliding sleeve provides the means of flow rate adjustment. This approach permits stoichiometry control on an individual burner basis.

The structural component most critical to the combustion process is the burner exit or quarl. Considerable variation in design is found in current boiler practice. The exit generally involves an expansion from the burner throat. The expansion may be conical with half-angles ranging from 20 to 60 degrees, or may be convex in shape. The exit section surface can be either refractory or watertubes. If it is refractory, it may have studded tubes buried within it. The exit section and the swirl generators function as a unit to achieve stable combustion.

Equipment is available for boiler units ranging from the smallest to the largest that allows the boiler operator to ignite the main fuel by the simple expedient of pressing a button. This equipment ranges from spark devices that ignite fuel oil directly, to gas or light oil equipment with spark ignition which is used for ignition of the main streams of gas and fuel oil. These devices are available with control equipment that ranges from the simplest push button requiring observation of ignition by the operator at the burner, to a fully "programmed" starting sequence, complete with interlocks and flame-sending equipment, all remotely operated from the boiler control room. For large steam generators, it is necessary to keep the igniters operating for hours prior to main fuel start-up until the temperature in the combustion zone and air preheat temperature become high enough to assure self-sustaining ignition of the main fuel.

The self-igniting characteristics of pulverized coal vary from one fuel to another, but on most coals, it is possible to maintain ignition without auxiliary fuel down to one-third capacity of the boiler. Below 50 percent boiler load, pulverizers may be removed from service to reduce power consumption and wear. In some instances, completely reliable ignition is obtained down to a quarter load. When firing pulverized coal with low volatile matter, it may be necessary to activate the igniters even at high loads. This can occur with any coal if it is excessively wet, frozen, and/or feeding sporadically into the pulverizers. If the igniter is not activated during the intervals when coal is not reaching the burner, or is reaching it in small amounts, ignition of the burner may be lost momentarily; then upon reestablishing coal flow, the burner may puff or ignite with explosive force from an adjacent burner.

One of the most important items required in a burner control system is a means of determining the presence or absence of flame on each individual burner regardless of the type of fuel being burned. Ultraviolet (UV) flame detectors have been applied to all types of fuels. The UV detector is especially well-suited to natural gas-fired burners due to the abundance of ultraviolet radiation produced by the combustion of the hydrogen in natural gas. A detector has been developed which uses the high frequency dynamic flickering of the primary combustion process in combination with the intensity of the visible radiation of the flame. Application of this dual-signal detector indicates that the new "flicker" detector provides greater discrimination and reliability than the UV detector on oil and coal-firing. The two detectors can be used in combination on a burner where it is necessary to monitor both gas and oil or coal-firing on the same burner, such as on a coal burner with a gas lighter. Another type of scanner has its sensitivity in the infrared portion of the spectrum, and may also be combined with a frequency or "flicker" type of discriminator.

3.3.2 Low NO_x Burners

The major design criteria for pre-NSPS burners were the optimization of flame stability and combustion efficiency. For the high intensity standard burners discussed above, these objectives were effectively achieved by providing for rapid mixing of the fuel and air to produce compact, intense flames. However, these conditions also favor the formation of NO_x. The basic premise behind the design of low NO_x burners is to meet the NSPS for NO_x while satisfying the performance criteria established for pre-NSPS burners. The following methods are utilized to reduce NO_x emissions:

- Lower combustion zone temperature;
- Low excess air firing (where fuel properties are acceptable);
- Fuel/air mixing schedule control;

The combustion zone temperature is usually lowered by increasing the furnace surface area and by using burners with less turbulent flames. This reduces the flame intensity and distributes the heat release over a larger region. Low NO_x burners are able to produce less turbulent flames by controlling fuel/air mixing rates. Separation of the secondary airstream into two channels is also commonly used to delay mixing of the fuel and oxidizer. These methods of lowering the flame temperature are preferred to simple techniques such as reducing the air preheat temperature. This is because lower air preheat influences primarily thermal NO_x formation while the methods described above involving fuel/air mixing affect both thermal and fuel NO_x formation. Combinations of mixing controls and lower preheat are not used since flame stability and carbon burnout may be unacceptable.

Low excess air firing is not usually considered a burner design variable, but it may exert an influence of the design process. For instance, a major factor in low excess air firing is balancing the airflow between burners. This requires a complex system of dampers, valves, flowmeters, and registers (or their equivalent) which may be incorporated into the burner design or included in the system, exterior to the burner. Low excess air firing not only affects NO_x by providing fewer oxygen molecules for combination with nitrogen, but it increases the process efficiency reducing enthalpy losses to the stack gas. Obviously, lowering the excess air too far will cause increased CO emissions and carbon in the ash. This explains the importance placed on proper distribution of the air between burners. The desired fuel/air mixing schedule determines the most noticeable features of low NO_x burner designs. As discussed above, standard pulverized coal burners usually have one annular secondary air passage which surround a central core. Low NO_x burners have two concentric annular secondary air passages. Splitting the secondary air delays mixing of the primary and secondary airstreams and provides mixing control by allowing different velocities and/or swirl components in the two channels.

Foster Wheeler, Babcock and Wilcox, and other burner manufacturers have designed and tested low NO_x burners. The Foster Wheeler design is called the "controlled flow burner" and utilizes an annular coal nozzle with a tangential inlet to provide primary swirl. The Babcock and Wilcox design is called the "dual register burner" and has a central coal nozzle with a venturi to produce a uniform distribution of coal across the nozzle exit. The design and performance of these burners in field boilers are discussed in Appendix D.

3.4 Comparison of General DMB Design Requirements with Standard Commercial Practice

Table 3-7 compares the general DMB design requirements (Table 3-5) with standard commercial burner practice as discussed in the previous sections. For most of the items listed, the general DMB design requirements will require minor modifications to commercial practice. Exceptions are discussed below.

The most important difference between the distributed mixing burner and commercial practice is the tertiary air ports. These ports will require additional penetrations through the furnace tube-wall and must incorporate an airflow control system. In the distributed mixing burners tested previously at EER, the airflows to each burner passage were measured with venturis and controlled by gate or butterfly valves. However, in commercial practice all of the combustion air (except for the primary) flows through a windbox. Airflow to the various DMB passages must, therefore, be controlled by a different method such as dampers or other control devices within the windbox.

The air velocities through the primary and secondary channels are lower in the distributed mixing burner than in commercial practice. This will require a larger fuel nozzle. The secondary air passage and burner throat may not need to be enlarged because although the velocities are lower, a significant portion of the combustion air is diverted through the tertiary air ports.

Table 3-7. Comparison of Prototype Burner Design Criteria with Standard Commercial Practice

SYSTEM/COMPONENT	STANDARD COMMERCIAL PRACTICE	PROTOTYPE BURNER DESIGN CRITERIA
• Fuel Injector		
– Percent of total air	10-40%	20%
– Injection velocity at max load	75-90 fps	75 fps
– Minimum injection velocity	50 fps	50 fps
– Mechanical spreader	Used by some manufacturers	Depends on desired flame shape
• Fuel System		
– Flow control and measurement	Feeder speed	Feeder speed
– Multifuel capability	Gas, oil, wood, COM	Same as host site
– Turndown range	3-4	3
• Combustion Air Systems		
– Primary velocity	50-90 fps	75 fps at max rating
– Secondary velocity	~100 fps at max rating	60 fps at max rating
– Secondary swirl	High	Low-medium
– Excess air	15-30%	~15%
– No. of secondary channels	One or two	Two
– No. of tertiary ports	Not applicable	Four
– Secondary airflow control and measurement	Registers for control, no individual burner measurement *	Integrated control and measurement (see text)
– Windbox	Compartmented on a few new units, otherwise open	Open
– Overfire air	Some units	None
• Burner Exit		
– Half angle	20-60°	25°
– Construction	Tubes, ceramic w/ or w/o studded tubes	Tubes, ceramic w/ or w/o studded tubes
• Lighter		
– Fuel	Oil, gas electric	Same as host site
– Duty	Continuous, on during load changes	Continuous, on during start-up
• Control System		
– Ignition sequence	Manual or automatic	Manual
– Response to steam demand	Automatic	Automatic
• Safety System		
– Main fuel shutoff	Automatic or manual based on lighter and/or main flame	Automatic based on lighter or main flame
– Type of flame detection	or flicker	Same as host site

* Note that the Foster Wheeler low NO_x burner incorporates flow rate measurement on an individual basis.

The burner exits for the distributed mixing burners tested to date have been considerably larger than those of standard commercial burners. In field boilers, the burner exit length and outside diameter are limited by the furnace and windbox designs. Distributed mixing burners for field boilers must conform to these conditions.

3.5 Integrated Prototype Burner Designs

The protototype burners must be compatible with the specific characteristics of the host boilers and their ancillary equipment and be capable of operating under the same conditions as the burners currently installed in the host boilers. The operational aspects of the prototype burners will be evaluated during tests at EER. The discussion in this section is limited to the host boiler parameters which influence the prototype burner design. These include:

- Fuel Supply System
    - Coal composition
    - Mill air requirements (primary stoichiometry)
    - Primary air temperature
    - Pulverizer out of service operating requirements
- Air Supply System
    - Overall excess air
    - Air heater performance (windbox air temperature)
    - Fan capacity (windbox pressure)
    - Windbox dimensions
- Boiler/Furnace
    - Burner capacity
    - Furnace depth
    - Burner array configuration
    - Burner-to-burner spacing
    - Structural member locations

Most of these parameters impact the prototype burner design by affecting the actual flow rates of air required through each burner passage. These flow rates combined with the velocities specified in the distributed mixing burner design criteria, allow the areas of each burner passage to be calculated.

The windbox depth is the maximum axial distance available for the burner exit and registers, and thus, influences the maximum possible burner exit length. This maximum length is generally smaller than the optimum, as specified by the distributed mixing burner design criteria, and will thus require some compromises.

The furnace depth determines the maximum possible flame length. As previously discussed, the coal nozzle design can be varied to adjust flame shape and length. Short furnace depths will require a coal nozzle design which will achieve rapid mixing of the primary and secondary combustion air. For longer furnace depth, the coal nozzle should be designed to provide slower mixing.

The burner array configuration, burner-to-burner spacing and the location of the tube-wall and windbox structural elements will affect the tertiary port locations. The largest possible distances from the burner centerline to the tertiary port centerlines are achieved by combining the tertiary ports for adjacent burners midway between the burners. This was the method utilized in the four-burner tests at EER, and results in a minimum number of discrete tertiary ports. In the candidate host boilers, this maximum distance is less than the distributed mixing burner design criteria, and thus, is more desirable than alternate tertiary port locations which would result in smaller distances. The tertiary ports must also be located to minimize interference with windbox and tube-wall structural elements (buckstays and trusses). This may require further compromises on tertiary port location.

3.5.1 Industrial Boiler Prototype Burner

The industrial prototype burner was designed to achieve the DMB design point velocities at a burner firing rate of 80×10^6 Btu/hr. This corresponds to the MCR burner capacity for a candidate host boiler located in South Carolina. At the time the prototype burner was designed, the South Carolina unit was the top candidate for the initial demonstration site. However, subsequent developments led to its elimination and the selection of the Illinois unit as the initial demonstration site. The normal burner duty cycle at the Illionois unit involves burner firing rates as low as 40×10^6 Btu/hr and as high as 100×10^6 Btu/hr with a MCR rating of 69×10^6 Btu/hr and a four-hour peak rating of 80x10^6 Btu/hr. Burner firing rates greater than 80x10^6 Btu/hr are used with two burners out of service. Since the prototype burner design point falls within the operating range of the initial demonstration site. it will be initially tested as designed. However, depending on the test results, it may be modified to achieve the DMB design point velocities at a lower load. This will require modifying the burner exit and sleeving the secondary and tertiary air passages to reduce their open area.

Figure 3-18 is a cross-sectional view of the prototype burner showing its design details. The following Foster Wheeler commercial burner components have been incorporated into the design:

- Annular Coal Nozzle;
- Telescoping Coal Nozzle for Primary Velocity Control;
- Registers for Swirl Control and Flow Shutoff;
- Airhoods for Secondary Flow Rate Control;
- Commercial Flame Scanning and Ignition System;

The coal and primary air enter through a tangential inlet to provide swirl. The coal nozzle is a tapered annulus, and the axial velocity increases as the coal and air move toward the burner throat. The exit end of the coal nozzle is formed from two concentric cones. The inner cone can be moved axially to vary the exit area, and hence, the primary velocity. The outer portion of the coal nozzle is constructed of removable sections so that the overall length of the coal nozzle may be changed to vary the coal nozzle set back within the burner throat. There are two concentric secondary air passages supplied from a common windbox. Each is equipped with an adjustable register for swirl control, an axially movable sleeve for flow rate control, and a perforated plate (air hood) for flow rate measurement. Pressure taps on both sides of each perforated plate allow the pressure drop to be measured and correlated with sleeve position and flow rate.

The portion of the burner inside the coal nozzle is termed the core, and is supplied with a small amount of cooling air from the windbox. A retractable ignition system, including spark ignition, a 10x10^6 Btu/hr oil nozzle and a pilot flame scanner are located in the core on the burner centerline. The main flame scanner views the flame through the inner secondary air passage.

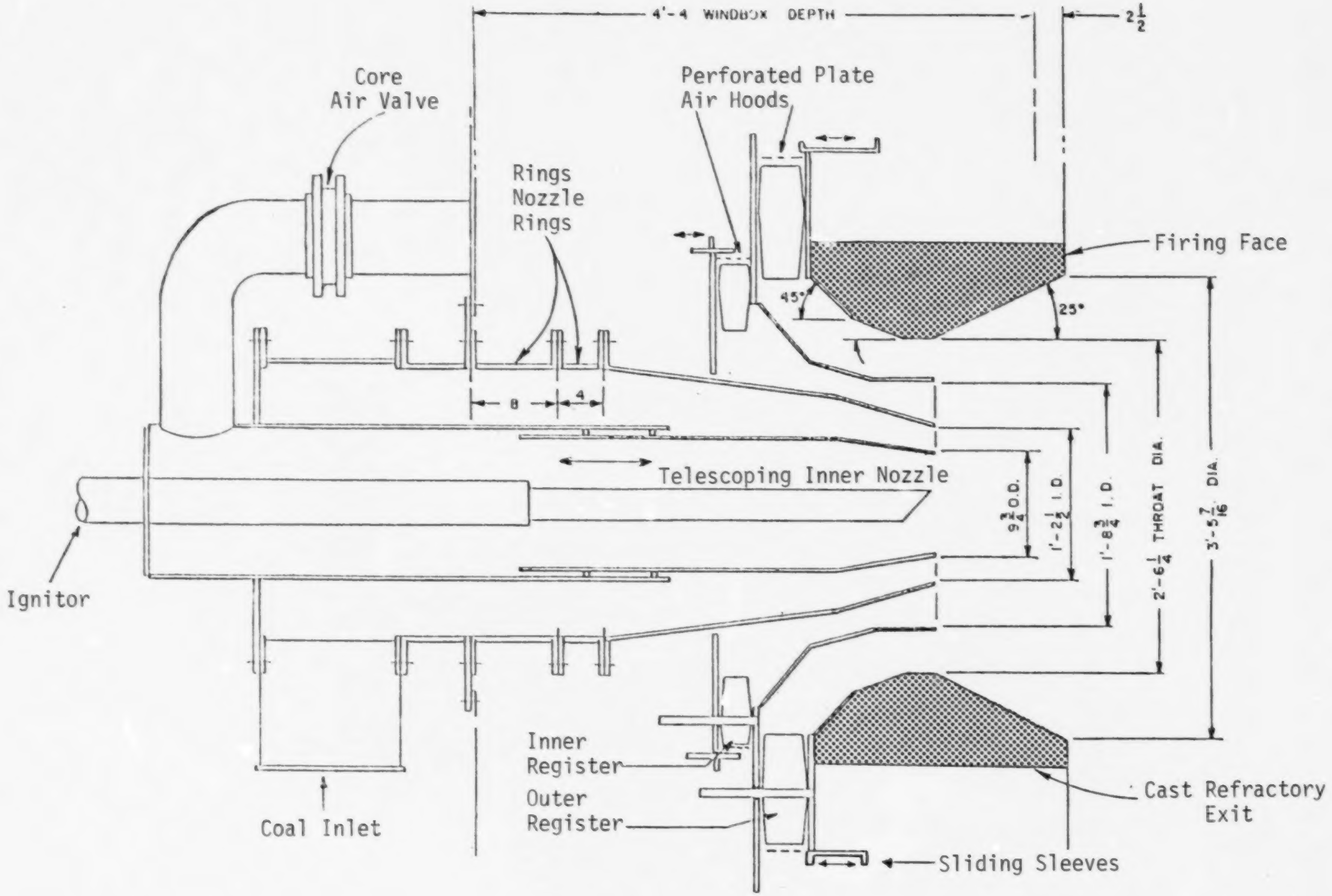

4'-4 WINDBOX DEPTH
2 1/2
Core Air Valve
Perforated Plate Air Hoods
Rings
Nozzle
Rings
Firing Face
45°
25°
8
4
Telescoping Inner Nozzle
9 3/4 O.D.
1'-2 1/2 I.D
1'-8 3/4 I.D
2'-6 1/4 THROAT DIA.
3'-5 7/16 DIA.
Ignitor
Inner Register
Outer Register
Coal Inlet
Cast Refractory Exit
Sliding Sleeves

Figure 3-18. Cross-Section of Industrial Prototype Burner Based on Foster Wheeler Design, Without Tertiary Air Ports.

Tertiary air is supplied from the windbox through outboard tertiary air ports. The airflow rate through each port will be controlled and measured by the assembly shown in Figure 3-19. The hemispherical poppets pivot to vary the open area and thus control the flow rate. The pressure drop across the perforated plate provides an indication of flow rate. Four tertiary air ports will be used for the prototype burner tests. Their locations with respect to the burner throat are shown in Figure 3-20. The tertiary port spacing matches the burner spacing in the host boiler. The port location to be used in the field will be selected to uniformly distribute the tertiary air around and between the burners. Some compromise may be required depending on windbox construction and furnace structural considerations.

In summary, the prototype burner has been designed with "flexible" parameters to permit the flow rates, velocities and swirl in the burner passages to be varied to optimize burner performance. The dimensions and ranges of adjustment are listed in Table 3-8. All of the dimensions listed match the prototype design point except the exit length which must be considerably shorter than the design point due to windbox depth limitations. The resulting exit-length-to-throat-diameter ratio is 0.40 compared to 1.0 for the DMB design point. The effects of this shorter exit length will be evaluated during the prototype tests.

3.5.2 Utility Boiler Prototype Burner

Since a utility host site has not been selected, a preliminary prototype burner has been designed based on the approximate utility boiler demonstration site characteristics listed in Table 3-3. This burner design is preliminary in that it represents the general burner design and does not include the flexibility inherent in the industrial prototype burner. If the industrial prototype burner test results show that additional flexibility is required in the utility prototype burner, it will be incorporated into the final design.

Figure 3-21 is a cross-sectional view of the preliminary utility prototype burner showing its design details. The following Babcock and Wilcox commercial burner components have been incorporated into the burner.

- Axial coal nozzle;
- Registers for flow shutoff;

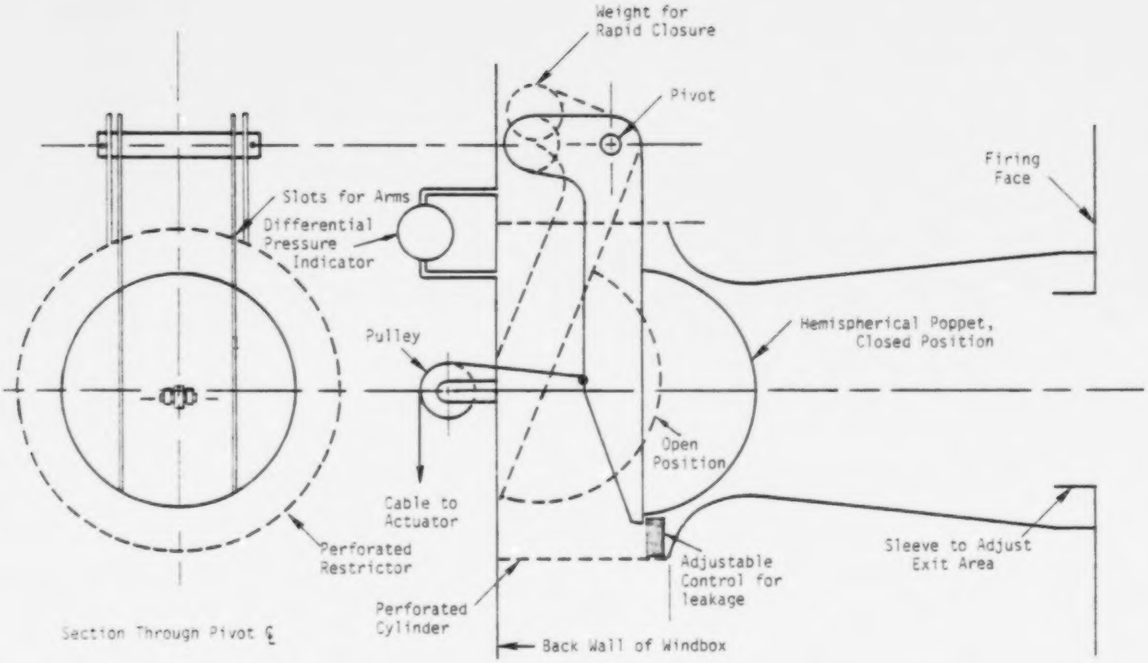

Weight for Rapid Closure
Pivot
Firing Face
Slots for Arms
Differential Pressure Indicator
Pulley
Hemispherical Poppet, Closed Position
Open Position
Cable to Actuator
Perforated Restrictor
Adjustable Control for leakage
Sleeve to Adjust Exit Area
Perforated Cylinder
Section Through Pivot ℄
Back Wall of Windbox

Figure 3-19. Tertiary Control Valve.

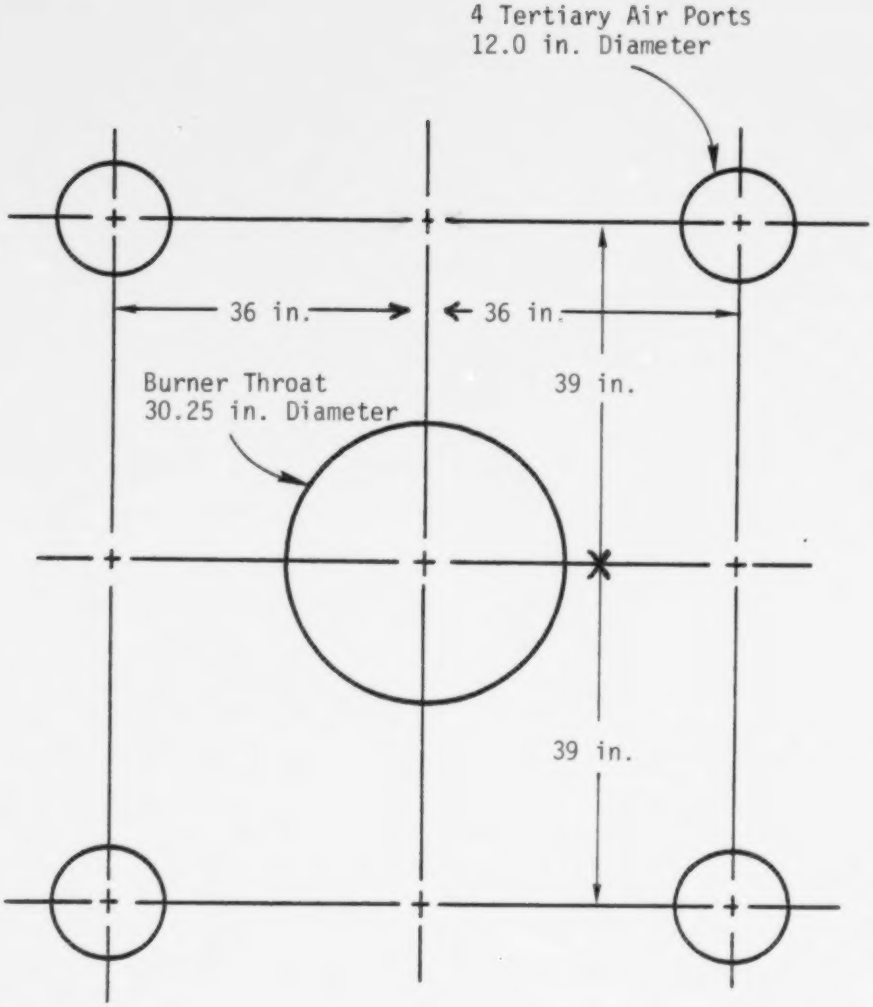

4 Tertiary Air Ports
12.0 in. Diameter
36 in.
36 in.
39 in.
Burner Throat
30.25 in. Diameter
39 in.

Figure 3-20. Tertiary Port Arrangement for Industrial Prototype Burner.

Table 3-8. Industrial Prototype Burner Dimensions and Adjustments

		Adjustment	
Design Variable	Nominal Value	Range	Method
Fuel Injector			
Core Diameter (in.)	9.75		
Core Area (in.^{2})	74.7		
Nozzle Diameter (in.)	14.5		
Annulus Thickness (in.)	2.375	1.7 to 3.3	Telescoping Nozzle
Primary Area (in.^{2})	90.5	68 to 136	
Setback (in.)	0	-2 to +12	Remove/Replace Nozzle Rings
Inner Secondary Channel			
Outer Diameter (in.)	20.75	Variable	
Annulus Thickness (in.)	3.125	Variable	Replace Secondary Divider
Area (in.^{2})	173.0	Variable	
Setback (in.)	0	Variable	
Outer Secondary Channel			
Outer Diameter (in.)	30.25	Variable	
Annulus Thickness (in.)	4.75	Variable	Recast Exit
Area (in.)	380.5	Variable	
Setback (in.)	0	Variable	
Tertiary Ducts			
Distance from Burner ℄ (in.)	53	Variable	Modify Windbox
Spacing Around Burner (degrees)	90	Variable	
Number of Ports	4		
Injection Angle (degrees)	0		
Axial Position (in.)	0		
Diameter (in.)	13.25	Variable	Sleeve
Total (in.^{2})	553	Variable	
Throat and Exit			
Thrpat Diameter (in.)	30.25	Variable	Recast Exit
Throat Area (in.^{2})	718.7	Variable	
Half Angle of Exit (degrees)	25	Variable	
Length of Exit (in.)	12.0	Variable	
Length/Diameter	0.40	Variable	

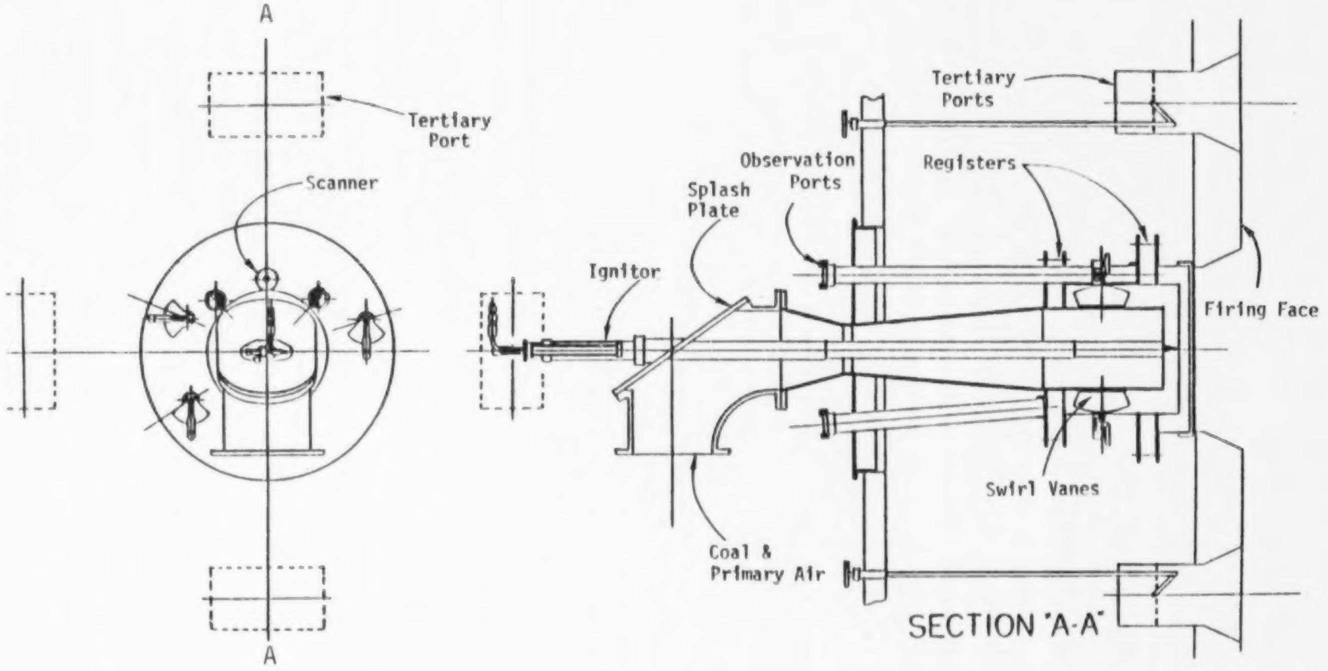

A
Tertiary Port
Scanner
A
Ignitor
Splash Plate
Observation Ports
Tertiary Ports
Registers
Firing Face
Swirl Vanes
Coal & Primary Air
SECTION 'A-A'

Figure 3-21. Cross-Section of Preliminary Utility Prototype Burner Based on Babcock and Wilcox Design.

- Swirl vanes for inner secondary swirl;
- Commercial flame scanning and ignition system;

The coal and primary air enters the nozzle through a 90^{o} elbow. The inlet end of the nozzle has a venturi which is designed to produce an approximately uniform distribution of coal at the nozzle exit. There are two concentric secondary air passages each equipped with a register which allow the airflow to be shut off when the burner is out of service. The outer register can be adjusted to provide swirl control. However, this will also vary the flow distribution among the various burner passages. The inner secondary is equipped with swirl vanes so that the inner secondary swirl is approximately independent of register position. There are four tertiary air ports with separate dampers on each. The tertiary ports are rectangular with the shorter dimension horizontal. This results in less disturbance to the tube-wall, and hence, greater structural strength than circular ports of the same area. The oil igniter is on the burner centerline and has a ceramic exterior to minimize wear. The scanner (not shown) views the flame through the inner secondary air passage.

The dimensions of this burner are listed in Table 3-9. As with the industrial prototype burner, the ratios of tertiary-port-radius-to-throat-diameter and exit-length-to-throat-diameter are less than the design point criteria.

Since the utility host boiler has not been selected at this time, it is not possible to specify the furnace firing depth, and hence, the maximum acceptable flame shape. Thus, it is desirable to provide some means of varying flame shape and length in the prototype burner. Previous DMB tests have shown that coal nozzle design is a key parameter influencing flame shape and length. Nozzles which channel the coal into an axial jet tend to produce long narrow flames, while nozzles which mix the coal more rapidly with the combustion air produce shorter flames. The utility prototype burner nozzle design should thus produce a long narrow flame.

Recent LWS tests of a subscale (i.e., 50×10^{6} Btu/hr) version of the Babcock and Wilcox Dual Register Burner (DRB) which is similar in construction to the preliminary prototype burner have provided the opportunity to evaluate the effects of nozzle design on flame shape. The design of this DRB is discussed in Appendix D. Both the DRB and prototype burner have: a central coal nozzle

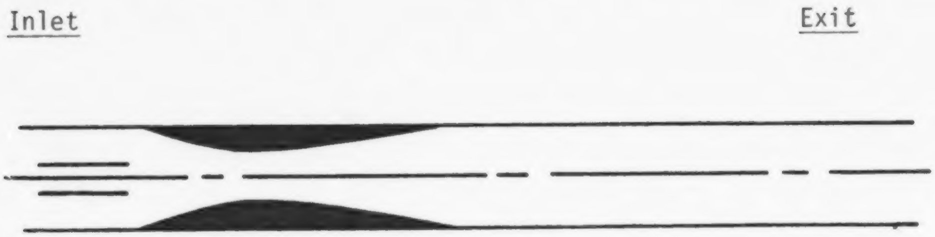
Table 3-9. Preliminary Utility Prototype Burner Dimensions

Design Variable	Nominal Value
Fuel Injector	
Nozzle Diameter (in.)	15.25
Primary Area (in.^{2})	182.6
Setback (in.)	3.0
Inner Secondary Channel	
Outer Diameter (in.)	25.0
Annulus Thickness (in.)	4.875
Area (in.^{2})	308.0
Setback (in.)	5.0
Outer Secondary Channel	
Outer Diameter (in.)	32.0
Annulus Thickness (in.)	3.5
Area (in.^{2})	313.4
Setback (in.)	5.0
Tertiary Ducts	
Distance from Burner ℄ (in.)	48.0
Spacing Around Burner (degrees)	90
Number of Ports	4
Axial Position (in.)	0
Injection Angle (degrees)	0
Length x Width (in.)	22.0 x 12.0
Total Area (in.^{2})	1065.0
Throat and Exit	
Throat Diameter (in.)	32.0
Throat Area (in.^{2})	804.0
Half Angle of Exit (degrees)	25
Length of Exit (in.)	7
Length/Diameter	0.219

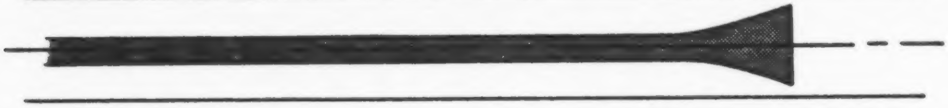
with a venturi mixing device; two concentric annular secondary air passages for secondary airflow rate control; and adjustable swirl vanes in the inner secondary air passage. Although the DRB was not designed for use with tertiary air ports, ports were installed so that the effects of staging could be evaluated. The design parameters for these burners are substantially different. The DRB has a capacity of 50×10^6 Btu/hr and the prototype burner will be approximately 125×10^6 Btu/hr. The velocities in the air passages are also different. In comparison to the prototype burner design point, the DRB primary and tertiary velocities are higher and the secondary velocity is lower. These differences in design parameters preclude the use of the DRB for detailed prototype burner tests. However, the similarities in the burner designs (and in particular, the coal injection schemes) should result in similar changes in flame shape and length with alternate coal nozzle designs.

The DRB was tested with the three nozzle designs shown in Figure 3-22. The baseline DRB mixing device is a venturi which is designed to produce a uniform distribution of coal and air in the primary nozzle. As expected, the flame produced with this nozzle design was long and narrow. Flame impingement was observed on the furnace rear wall which was set for a 16 foot firing depth. Two alternate mixing devices were also tested: a divergent cone which channeled the primary coal and air into an annulus, and the same cone with an impeller upstream to produce swirl. Both devices could be positioned at any axial location in the nozzle. The divergent cone did not significantly affect flame shape at any axial position. When the cone was positioned far from the burner throat, CO and NO emissions were comparable to those measured with the baseline venturi mixing device. When the cone was positioned near the throat, CO emissions were higher and NO emissions were lower than those measured with the baseline mixing device. This was probably due to the higher axial primary velocity in the annulus produced by the cone.

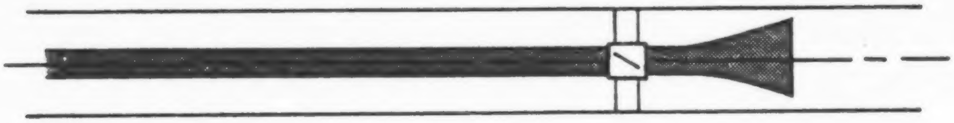
The impeller and cone mixing device was found to significantly influence flame shape. When this device was located near the throat, a short bright intense flame was produced, CO and smoke were lower than baseline and NO was higher than baseline. Axial position was found to significantly alter flame shape and the shortest flame was produced with the device nearest the burner throat. This mixing device was also tested with the tertiary ports operational (staged), as in the prototype burner, with similar results.



Baseline Venturi



Divergent Cone



Divergent Cone and Impeller

Figure 3-22. Coal Nozzle Designs.

In summary, these brief tests have shown that increased flexibility in shaping the flame can be obtained if additional mixing devices are included in prototype testing. This would allow greater flexibility in adapting the flame to meet the requirements of the eventual host boiler. The venturi and other mixing devices tested here provide for the ability to obtain a long narrow flame for applications in boilers having depths approaching 28 feet or greater, or a short wide flame (similar to that of the DMB) for boilers having depths of about 26 feet. These values are for single wall-fired units and must be increased for opposed wall-fired units. Based on these results, candidate mixing devices will be devised and evaluated for the utility prototype burner.

4.0 PREDICTION OF PROTOTYPE BURNER FIELD OPERATING CHARACTERISTICS

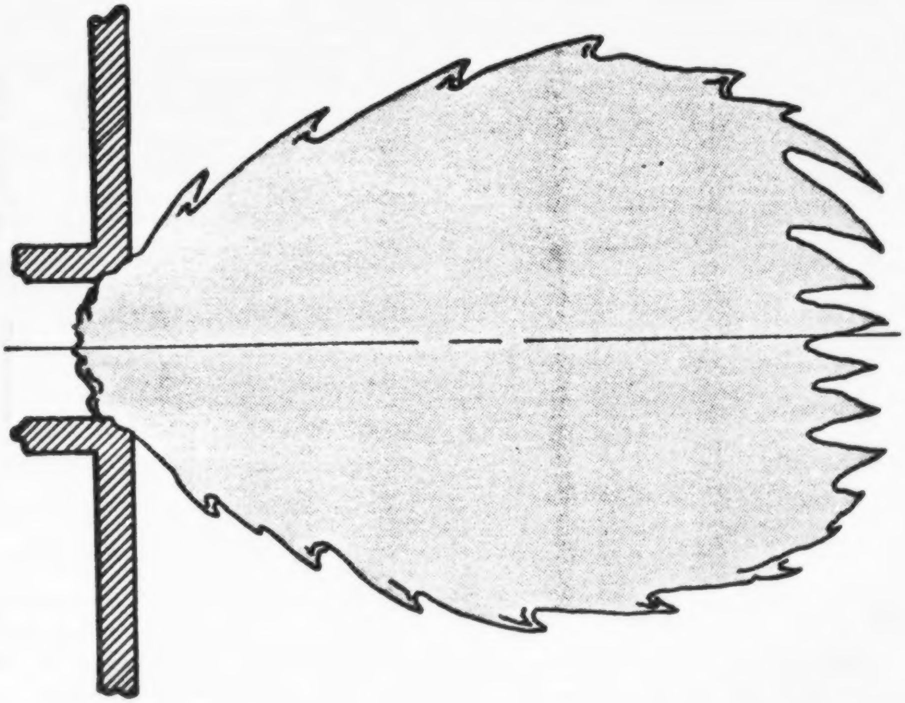
Since the basis for the prototype design is the research furnace (LWS) and not a steam raising system, some assurance is required that the DMB will perform in the field as predicted. Although indirect, the approach taken was to evaluate commercial burner operation in the LWS and compare its performance against field experience. This approach provides some assurance that if the burner operates satisfactorily in the LWS, it will also operate satisfactorily in the field.

In support of the Industrial Demonstration Program (EPA Contract 68-02-3127) a Foster Wheeler Intervane Burner (pre-NSPS) designed to meet the specifications of the small capacity host site boiler was evaluated. In addition, as part of the burner development program (EPA Contract 68-02-2667), a post-NSPS Babcock and Wilcox Dual Register Burner was also tested in the LWS. The design characteristics of these burners are discussed in Appendix D, and the results of the LWS tests are discussed and compared with field test data in the following two sections. The use of these data to predict prototype DMB performance in field boilers is discussed in Section 4.3.

4.1 Foster Wheeler Intervane Burner (Pre-NSPS)

The Intervane burner operational characteristics were evaluated in the LWS over a range in excess air and load. The burner installation and firing system were first checked by a qualified Foster Wheeler Service Representative to ensure that it conformed to practice. The service representative then supervised the burner start-up and adjustment of controls to establish the full load operating point providing acceptable flame shape/length and performance characteristics. At this burner setting, the input coal and airflows were then varied (identical to that in field operation) over the operating load range and at near full load over a range in excess air.

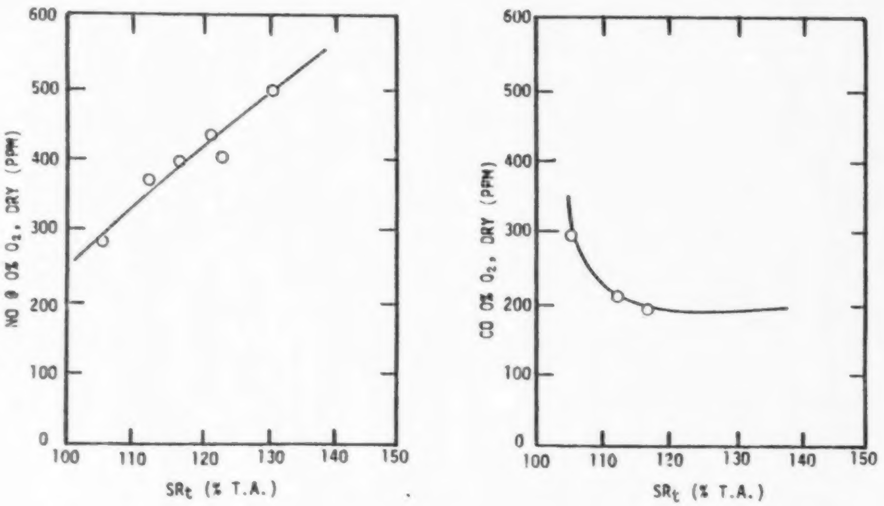
In general, the flames produced by this burner were short, bright, intense and stable. As shown in the sketch on the next page, the base of the flame was within the cylindrical portion of the exit and filled the divergent portion. In the furnace, the flame expanded rapidly and was about 12 feet (five throat diameters) long. The burner controls (register and core air control valve) affected the flame shape and optimum results were obtained with the register set at 40° open and the core air valve set at 78° open. These settings are similar to those used in field installations, but the flame shape in the LWS was somewhat



narrower and longer than those observed with multiburner arrays in the field. Throughout the tests, stack opacity was low except for low excess air conditions where some black smoke was observed. The smoke was generally a light gray haze. This similarity of flame characteristics between the LWS and field Intervane burner test results suggests that the short flame DMB flame characteristics and stability observed in the LWS will also be similar in the field.

The NO and CO characteristics for this burner are shown in Figure 4-1. The excess air results shown in Figure 4-1a are for a heat input of 80×10^6 Btu/hr and reveal that the NO level at 125 percent excess air is 450 ppm. (It should be noted that the MCR for this burner is about 90×10^6 Btu/hr which results in an NO level of 500 ppm.) These levels are 50 to 100 ppm lower than expected in a multiburner field unit of similar burner zone heat release rate; however, the variation in NO with excess air is within normal field observed ranges. The CO characteristics, also shown in Figure 4-1a, were more sensitive to variations in excess air than observed in the field. For example, in field units it is expected that the sharp rise in CO would occur at lower values

A. EXCESS AIR CHARACTERISTICS @ 80x10^6 Btu/hr



TEST CONDITIONS

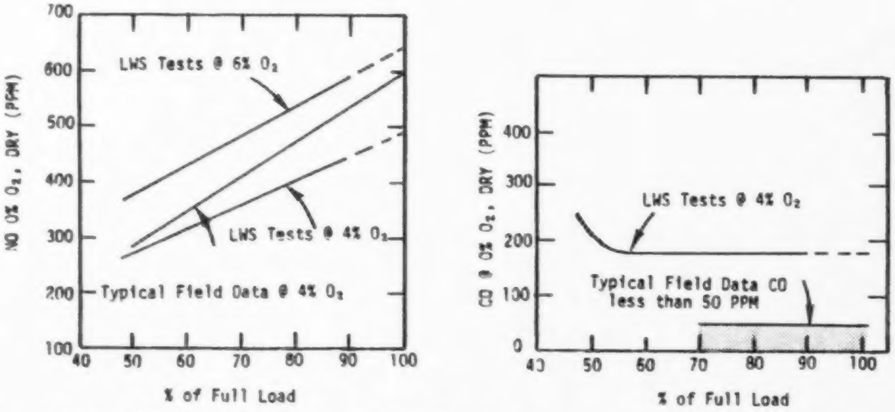
Register = 40° Open

Core Valve = 78° Open

SR_p = 18% T.A.

T_p = 160°F

T_s = 600°F



TEST CONDITIONS

Register = 40° Open

Core Value = 78° Open

T_p = 160°F

T_s = 600°F

SR_p = Follows Field Condition

B. LOAD CHARACTERISTICS AND FIELD DATA

Figure 4-1. Performance/Emissions Characteristics for Foster Wheeler Intervane Burner.

of excess air (<110 percent T.A.) than that shown in Figure 4-1a. In addition, the asymptotic value of CO measured in the LWS was about 185 ppm (corrected to 0 percent O_2) which is greater by a factor of four than that observed in multi-burner field units.

The effects of load on NO/CO emissions are shown in Figure 4-1b. The load was varied from about 45×10^6 Btu/hr heat input to about 80×10^6 Btu/hr. Based on an MCR of 90×10^6 Btu/hr, this represents a variation in percent of full load from about 50 percent to 90 percent. Over this range, the burner operation was stable; however, at the lower load steady-state coal feed rate was difficult to maintain. This problem was not related to the burner, but to the turndown capability of the C.E. pulverizer at the required coal-to-air ratios representative of Foster Wheeler type MB mills. Figure 4-1b also shows typical field data from an array of four Intervane burners of the same capacity (90×10^6 Btu/hr. The slope of the NO versus load characteristics for the field data is steeper than for the LWS test results. At the lowest load tested, 50 percent of full load, the field and LWS NO levels are almost identical. The CO levels measured in the LWS were nearly constant over the load range increasing about 50 ppm or 25 percent at the lowest load. The field CO data is also nearly constant, but substantially lower as discussed above.

4.2 Babcock and Wilcox Dual Register Burner (Post-NSPS)

The burner setup and checkout procedure for the Babcock and Wilcox Dual Register burner was identical to that described in Section 4.1. The general features of the dual register burner flame were considerably different from those of the high turbulence pre-NSPS Foster Wheeler Intervane burner. The flame was typical of long-narrow diffusion-type flames. The flame was, in fact, longer than the LWS furnace depth (16 feet), and this resulted in impingement of the flame on the back wall. As a consequence of this, incomplete combustion of the coal occurred resulting in rapid accumulation of residue in the bottom of the furnace and a very black (Ringleman number greater than about 3) exit plume.

The NO and CO characteristics for the Dual Register burner tested in the LWS are shown in Figure 4-2. The full load excess air characteristics (Figure 4-2a) show that CO increased as the excess air was decreased below 140 percent T.A. At a typical boiler excess air of 125 percent T.A., the CO level was

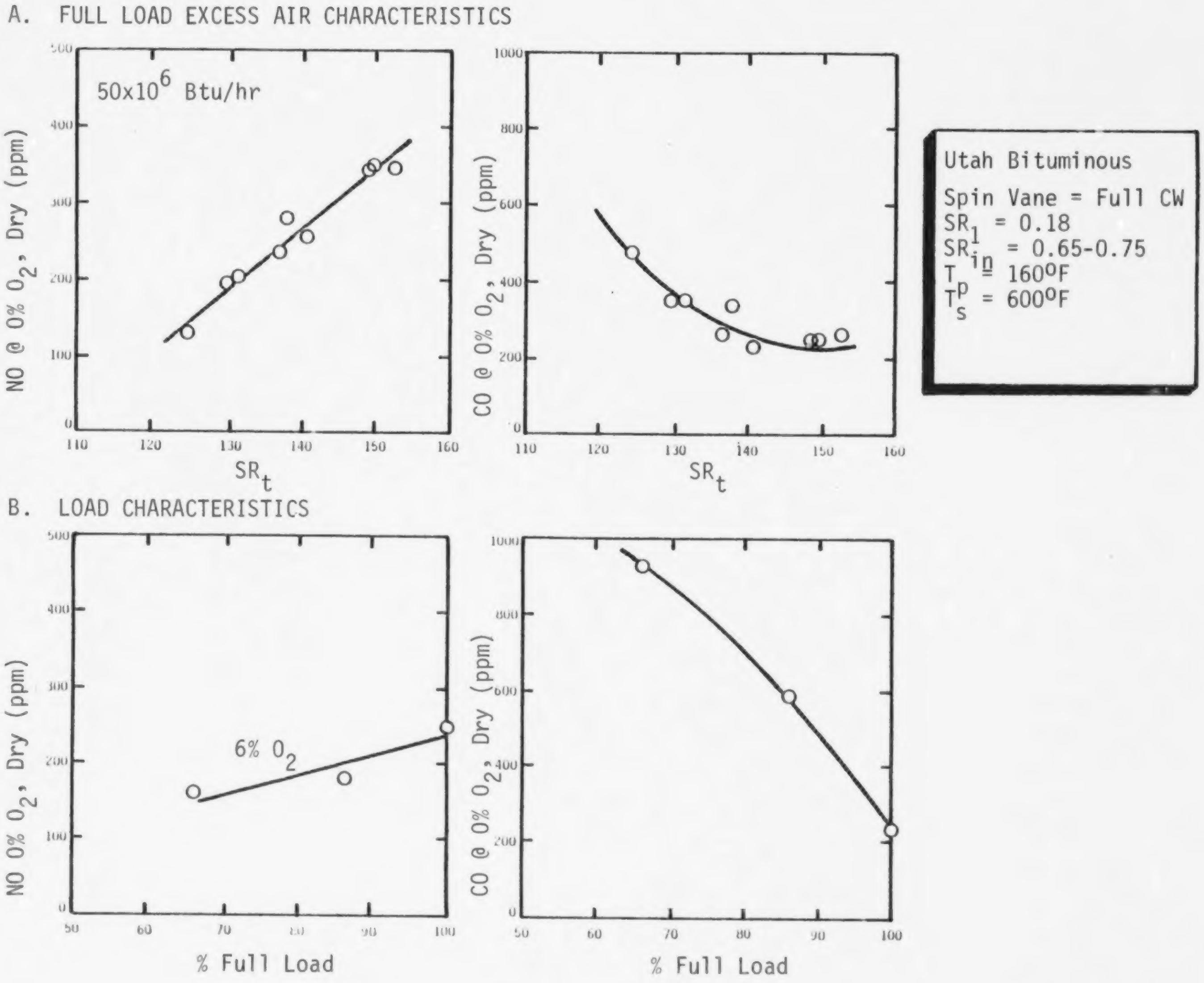


Figure 4-2. Performance/Emissions Characteristics for Babcock/Wilcox DRB in LWS - Single Burner.

500 ppm. CO also decreased as load was reduced even though the excess air was maintained high (6 percent O_2 or 138 percent T.A.). The table below compares the LWS test results with field data from a 500 MW opposed bituminous coal-fired boiler equipped with B&W Dual Register burners.

TABLE 4.1 EMISSION DATA COMPARISON

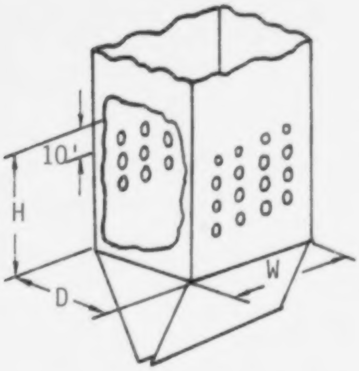
Load (% of full)	Excess O_2 (%)	EMISSIONS @ 0% O_2, DRY NO (ppm) LWS	NO (ppm) Field	CO (ppm) LWS	CO (ppm) Field
100	4.0	125	402	500	40
80	6.0	175	402	675	28

Due to impingement on the back wall, it is not surprising that these values deviate much further from field experience than the intervane burner LWS/field data comparison. EER is of the opinion that direct comparison of test results from impinging flame burners (such as the DRB in the LWS) with those from non-impinging flame burners (such as the DRB in typical field installations or the development DMBs in the LWS) is not possible because flame impingement significantly alters burner performance, the overall flame shape is altered, CO and unburned carbon losses are increased and NO is decreased. Therefore, no attempt will be made to predict DMB performance in field operating boilers based on the DRB test results in the LWS.

4.3 Projection of DMB NOx Levels to Field Conditions

Commercial burner manufacturers have found that the NO_x levels produced by the burners in boilers of their design can be approximately correlated with the total heat liberation per unit surface area in the furnace. Foster Wheeler's correlation uses a burner zone liberation rate (BZLR) defined as the net input to the burner zone divided by its effective projected surface (Ref 4.1). This surface, depicted in Figure 4-3a, is the area of the six-sided box starting at the hopper knuckle and extending 10 feet above the top row of burners, and includes all surfaces in that volume. Figure 4-3b shows the general trend of NO_x field test data for pre-NSPS Intervane burners in single wall-fired furnaces as a function of BZLR. The dotted curve below 125×10^3 Btu/hr ft^2 is an approximate extrapolation of field test results at higher BZLR. The BZLR for the LWS firing a 50×10^6 Btu/hr burner is 56×10^3 Btu/hr ft^2. Since this is in the

A. Surface Used for Correlation



B. Uncontrolled NO_x Emissions from Single-Wall Fired Units

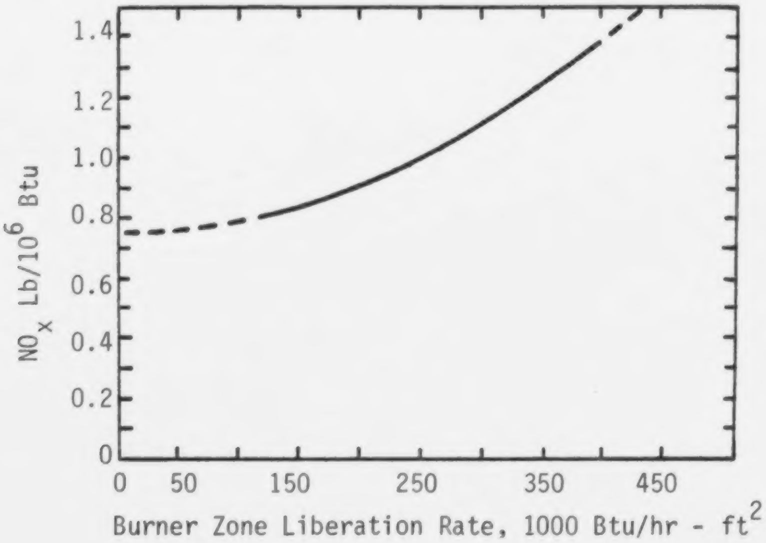


Figure 4-3. Foster Wheeler NO_x Emission Correlation.

extrapolation range and the LWS design is considerably different from Foster Wheeler furnaces (see Appendix B), this correlation cannot be used to accurately predict burner performance in field boilers. For example, based on the correlation, the Intervane burner (90×10^6 Btu/hr capacity) tested in the LWS should have a NO level of 0.75 lb/10^6 Btu or 590 ppm at 0 percent O_2 at a BZLR of 79×10^3 Btu/hr ft^2. However, the test results show an NO level of only 500 ppm which is considerably lower than predicted by the correlation.

Babcock and Wilcox also has a correlation relating NO_x emission (for their burners operating in boilers of their design) to heat release per unit surface area (HA/SC)(Ref 4.2). However, the Babcock and Wilcox correlation is based on a different definition of effective surface area. Figure 4-4 shows the results of field tests correlating NO_x emissions with HA/SC for pre-NSPS Babcock and Wilcox circular burners and the Babcock and Wilcox dual register burner. As with the Foster Wheeler correlation, the Babcock and Wilcox correlation was developed for their specific furnace designs. A majority of the data is at higher heat release rates and the results are extrapolated to lower heat release rate. Thus, it is not appropriate to apply this correlation to the DRB tests in the LWS. This is compounded by the flame impingement on the rear wall during the DRB tests in the LWS.

In addition to the BZLR correlation discussed above, Foster Wheeler has found that when two burners are tested in a research facility such as the LWS, the ratio of NO_x emissions measured in the test facility is approximately the same as the ratio of NO_x emissions from the same two burners operating in a commercial steam generator. Thus, the NO_x emissions from the DMB operating in a steam generator can be estimated from the following three sets of data:

- DMB operation in the LWS;
- Commercial burner operation in the LWS;
- Same commercial burner operation in a steam generator;

Of course this estimate would only be expected to be valid if the burner operated under comparable conditions in all three data sets. This eliminates the possibility of utilizing data from the DRB tests in the LWS since the flame impingement during the LWS tests is not typical of field operation. However, an estimate can be made based on the Intervane burner test data in Figure 4-1.

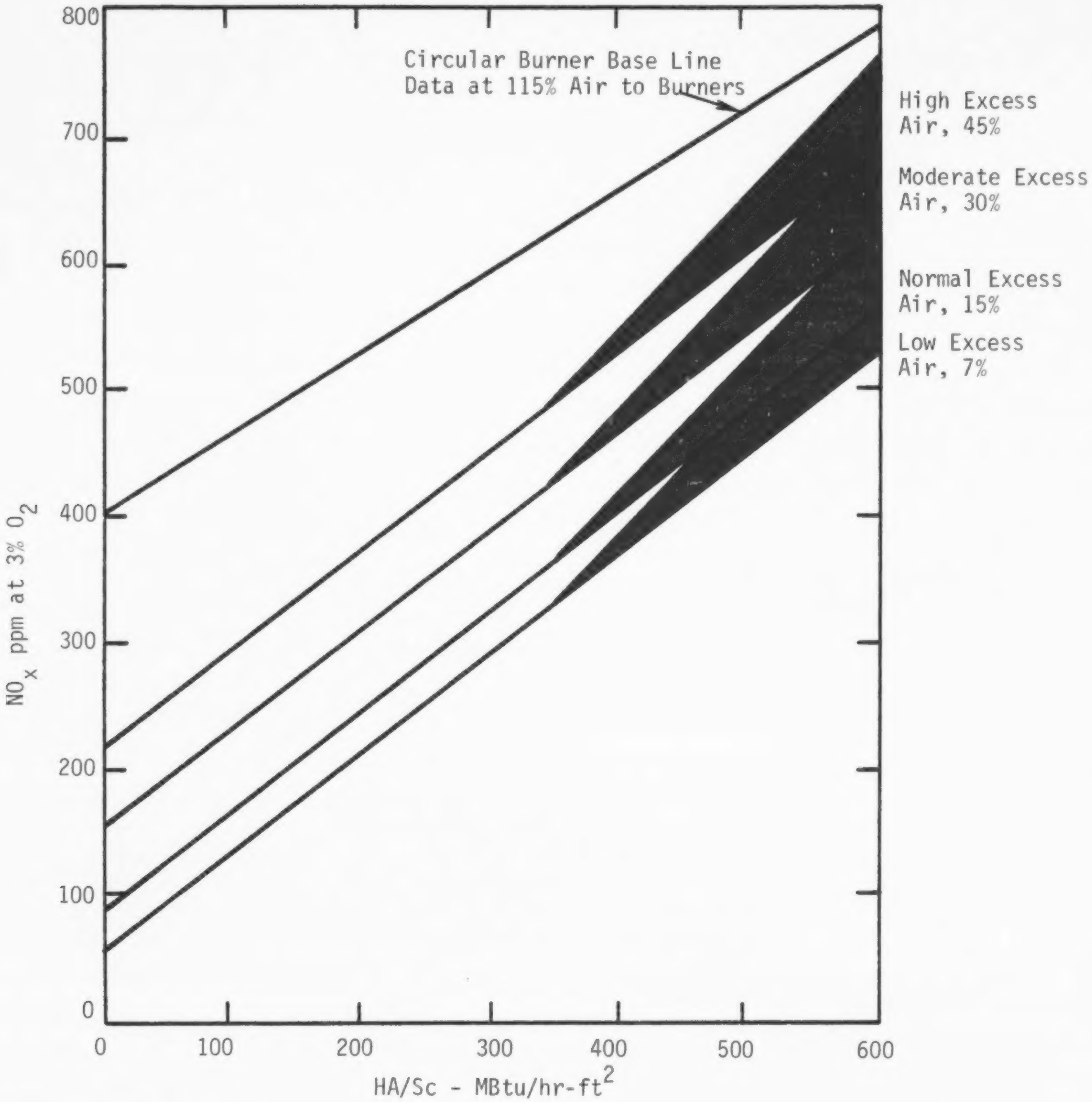


Figure 4-4. Babcock and Wilcox Correlation.

This estimate is summarized below:

TABLE 4.2. ESTIMATED EMISSIONS

Burner	LWS Tests		Field Operation	
	Capacity (10^6 Btu/hr)	NO @ 0% O_2 (ppm)	Basis	NO @ 0% O_2 (ppm)
Intervane	90	500	Typical Field Data	600
DMB	100	100	Predicted	≈120

The predicted NO level for the DMB operation in a steam generator (120 ppm) corresponds to 0.13 lb/10^6 Btu which is less than the program goal of 0.2 lb/10^6 Btu. However, there is considerable uncertainty in this estimate and the results should only be used as a rough indication of potential DMB performance in field units. A much more accurate estimate will be made by utilizing prototype DMB test data and baseline host boiler test data when available.

REFERENCES

4.1 Vatsky, Joel, "Experience in Reducing NO_x Emissions on Operating Steam Generators", Proceedings: Second NO_x Control Technology Seminar, EPRI Report FP1109-SR, July 1979.

4.2 Barsin, J.A., "Pulverized Coal Firing NO_x Control", Proceedings: Second NO_x Control Technology Seminar, EPRI Report FP1109-SR, July 1979.

APPENDIX A--DEVELOPMENT DISTRIBUTED MIXING BURNER DESIGN

Fundamental studies on NO formation in two-phase turbulent diffusion flames suggest the one method for minimizing NO_x emissions is to ensure that:

- The major fraction of the fuel nitrogen compounds contained in the coal are driven into the gas phase.
- The volatile fuel gases then experience an optimum stoichiometry history to provide for a maximum rate of decay of fuel nitrogen compounds to N_2. This stoichiometry will be very rich initially to prevent NO formation, but will require the addition of further oxygen to provide a fuel-rich mixture which minimizes the HCN content of the primary stage products.
- And lastly, ensure that sufficient enthalpy is lost from the initial heat release stages to prevent thermal NO production during the burnout phase.

The conceptual sketch shown below illustrates how the process and chemical requirements can be met. As shown, a key to achievement of the above requirements is to sequentially stage the fuel/air mixing in a distributed fashion.

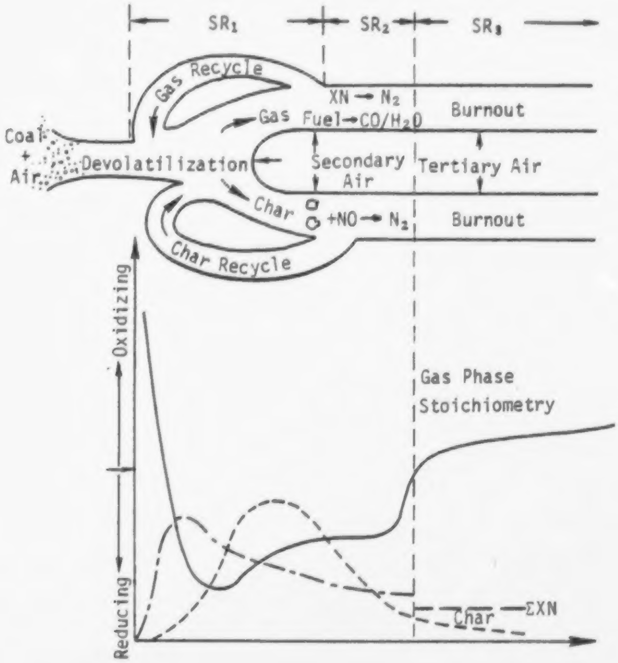

SR_1
SR_2
SR_3
Gas Recycle
$XN \rightarrow N_2$
Gas
$Fuel \rightarrow CO/H_2O$
Burnout
Coal + Air
Devolatilization
Secondary Air
Tertiary Air
Char
O
$C + NO \rightarrow N_2$
Burnout
Char Recycle
Oxidizing
Reducing
Gas Phase Stoichiometry
Char
ΣXN

FIGURE A-1. CONCEPTUAL SKETCH OF DMB.

Conceptually, this is accomplished by dividing the total input to the combustion chamber into three zones. In the first zone (SR_1) the stoichiometry is very fuel-rich, the objective being to prevent NO formation. Air is then added to the rich products (SR_2) to provide conditions which maximize XN decay. The region between SR_1 and the second dotted line shown in the above sketch represents the boundary of an internal recirculation zone whose intensity is not sufficient to allow complete penetration of the primary jet, thereby insuring that rapid mixing of the fuel and air in the initial two zones will not occur. The function of the remaining air (SR_3) is to provide further oxygen to ensure complete burnout, and also to provide a complete oxidizing zone around the rich primary zone. This prevents corrosion due to reducing atmospheres.

The DMB concept presented in Figure A-1 is one method that can be used to achieve the process requirements. Using this concept approach the design features which are used to provide controlled distributed fuel/air mixing are:

- The fuel injector design. The coal-plus-primary air is injected into the furnace either from a circular injector, an annulus, or from a divergent cone. The velocity is completely axial or injected with a swirling component provided either by swirl vanes in the primary jet or by introducing some tangential motion upstream of the fuel injector. These burner parameters provide highly uniformly mixed fuel with primary air to prevent the production of excessively-rich or excessively-lean zones;
- Swirl component of the secondary air. The swirl component in the secondary airstream may be adjusted to provide for the correct size and strength of the internal recirculation zone;
- Velocity ratio. The ratio between the secondary and primary streams controls the rate of mixing in the initial fuel-rich stream;
- Burner exit geometry. The rate of divergence of the burner exit controls the properties of the internal recirculation zone;
- Location of the outboard staged air jets and momentum. These dictate the residence time in the rich section as well as the time available for burnout;

In addition to providing those conditions which tend to minimize NO_x formation, all of these same parameters can be used to provide either a long flame or

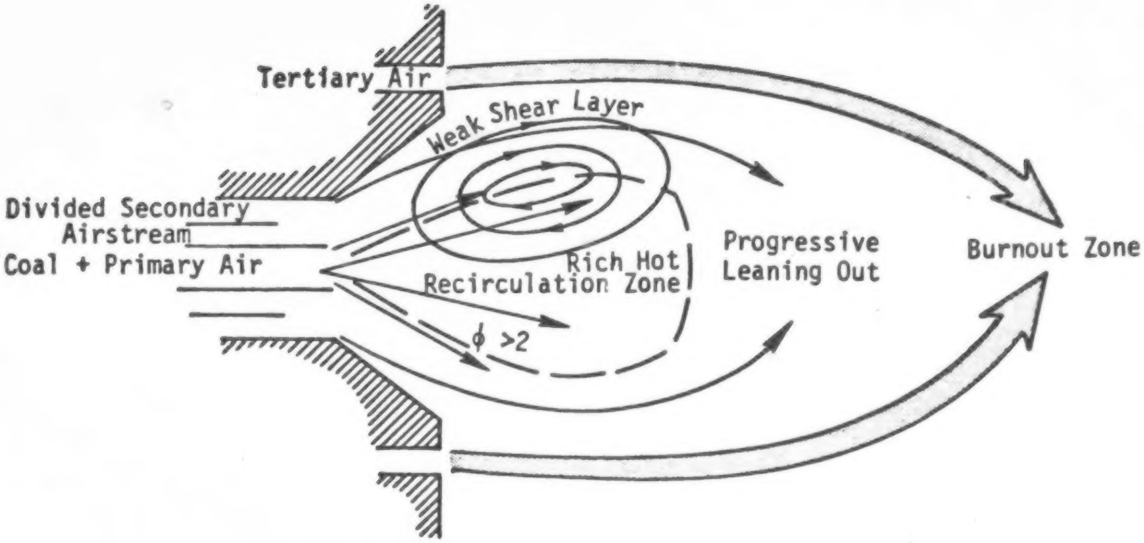

Tertiary Air
Weak Shear Layer
Divided Secondary Airstream
Coal + Primary Air
Rich Hot Recirculation Zone
ϕ >2
Progressive Leaning Out
Burnout Zone

Figure A-2. Distributed Fuel/Air Mixing Concept.

a short flame. The long flame is one method of minimizing NO by controlling the oxygen availability and maximizing residence time for conversion of XN to N_2. For many older conventional boilers, however, a long flame is not compatible with the available confinement; therefore, flexibility in adapting the flame to the shape required by the boiler furnace allows the optimization of the DMB concept in virtually all boilers. Also for new boilers, the long flame results in a limitation on the maximum burner liberation rate which results in an increase in the number of burners.

The DMB development design concepts can be categorized into two types:

1. Long diffustion flame - single and dual secondary burners.
2. Short flame - single secondary burners.

A total of five single burners and one multiburner array were evaluated. The burners are:

- Dual Secondary Burner
  - Single burner capacities

    $50x10^6$ Btu/hr
- Single Secondary Burner
  - Single burner capacities

    $10x10^6$ Btu/hr

    $12.5x10^6$ Btu/hr

    $50x10^6$ Btu/hr

    $100x10^6$ Btu/hr
  - Multiburner array

    2 x 2 array of $12.5x10^6$ Btu/hr burners (total capacity of $50x10^6$ Btu/hr)

A.1 Dual Secondary Burner

The dual secondary throat burner configuration is shown in Figure A-3. The major features of this burner are the following:

- Five independently-controlled airstreams divide the total combustion air into:

  ---the primary air carrying the coal;

  ---two concentric secondary airstreams to provide for inner and outer secondary airflow; and

  ---four outboard staged airstreams which can be divided into the upper or lower injectors.
- A central fuel injector with a retractable oil gun on the burner axis.
- The swirl level of the secondary airstream in either of the two channels is controlled by varying the air flow through the

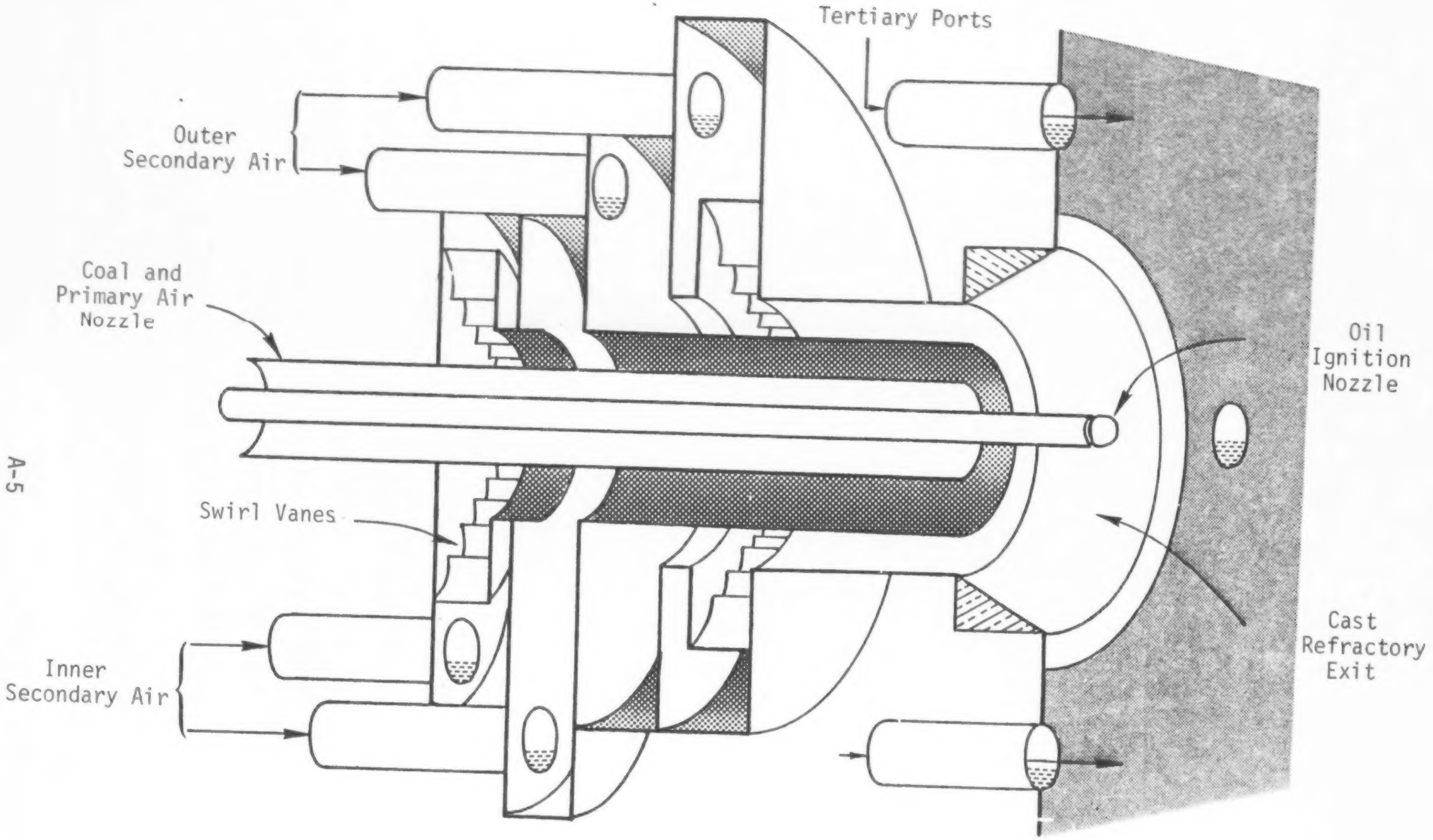

Tertiary Ports
Outer Secondary Air
Coal and Primary Air Nozzle
Oil Ignition Nozzle
Swirl Vanes
Cast Refractory Exit
Inner Secondary Air

Figure A-3. Dual Secondary Throat Distributed Mixing Burner (50×10^6 Btu/hr).

adjustable swirl vanes. Thus, the total swirl level in either channel can be varied from zero to a maximum dictated by the angle of the adjustable swirl vanes.

- Interchangeable ceramic burner exits.

A complete set of dimensions for the 50 x 10^6 Btu/hr burner is presented in Table A-1.

Figure A-4 presents an operating map for the 50 x 10^6 Btu/hr dual secondary burner showing the air velocities in the burner passages as functions of the burner zone (primary plus secondaries) and overall stoichiometries and load. These calculations were based on fixed primary stoichiometry and a low sulfur bituminous coal with a heating value of 12,340 Btu/lb. In the sample case indicated on Figure A-3 by "chase-around" lines and numbers, several operating conditions were first assumed. These were:

(1) Load = 50 x 10^6 Btu/hr

(2) Overall Stoichiometry = 150% T.A.

(3) Temperature of Primary Air = 150°F

(4) Burner Zone Stoichiometry = 60% T.A.

(5) Secondary Air Evenly Split Between Inner and Outer Ducts

With these five parameters fixed, all other burner flow parameters can be uniquely defined. (It should be noted that other combinations of parameters can also be chosen.

For this case the chart is entered (Lower Left Ordinate) at 50 x 10^6 Btu/hr and a horizontal line drawn completely across the figure. Where this line intersects the overall stoichiometry of 150 percent T.A., the temperature of primary air of 150°F and Burner Zone Stoichiometry of 60 percent T.A., the total Air Flow Rate ①, Primary Velocity ②, and Secondary Air Flow Rate ④ can respectively be determined. These values are:

Total Air Flow Rate = 12,500 Scfm

Primary Velocity = 72 ft/sec

Secondary Air Flow Rate = 7,000 ft^3/min

Table A-1. Design Information for Dual Secondary Burner.

Design Variable	Value
Fuel Injector	
Diameter (in.)	9.0
Area (in.^{2})	63.6
Swirl Angle (degrees)	None
Setback (in.)	0
Inner Secondary Channel	
Outer Diameter (in.)	15.5
Annulus Thickness (in.)	3.25
Area (in.^{2})	125.0
Setback (in.)	0
Outer Secondary Channel	
Outer Diameter (in.)	20.0
Annulus Thickness (in.)	2.25
Area (in.^{2})	125.0
Setback (in.)	0
Tertiary Ducts	
Distance from Burner Centerline (in.)	45.0
Spacing Around Burner (degrees)	90
Number of Ports	4
Injection Angle (degrees)	0
Axial Position (in.)	0
Diameter (in.)	9.0
Total Area (in.^{2})	254.0
Throat and Exit	
Throat Diameter (in.)	20.0
Throat Area (in.^{2})	314.0
Half Angle of Exit (degrees)	25
Length of Exit (in.)	18.5

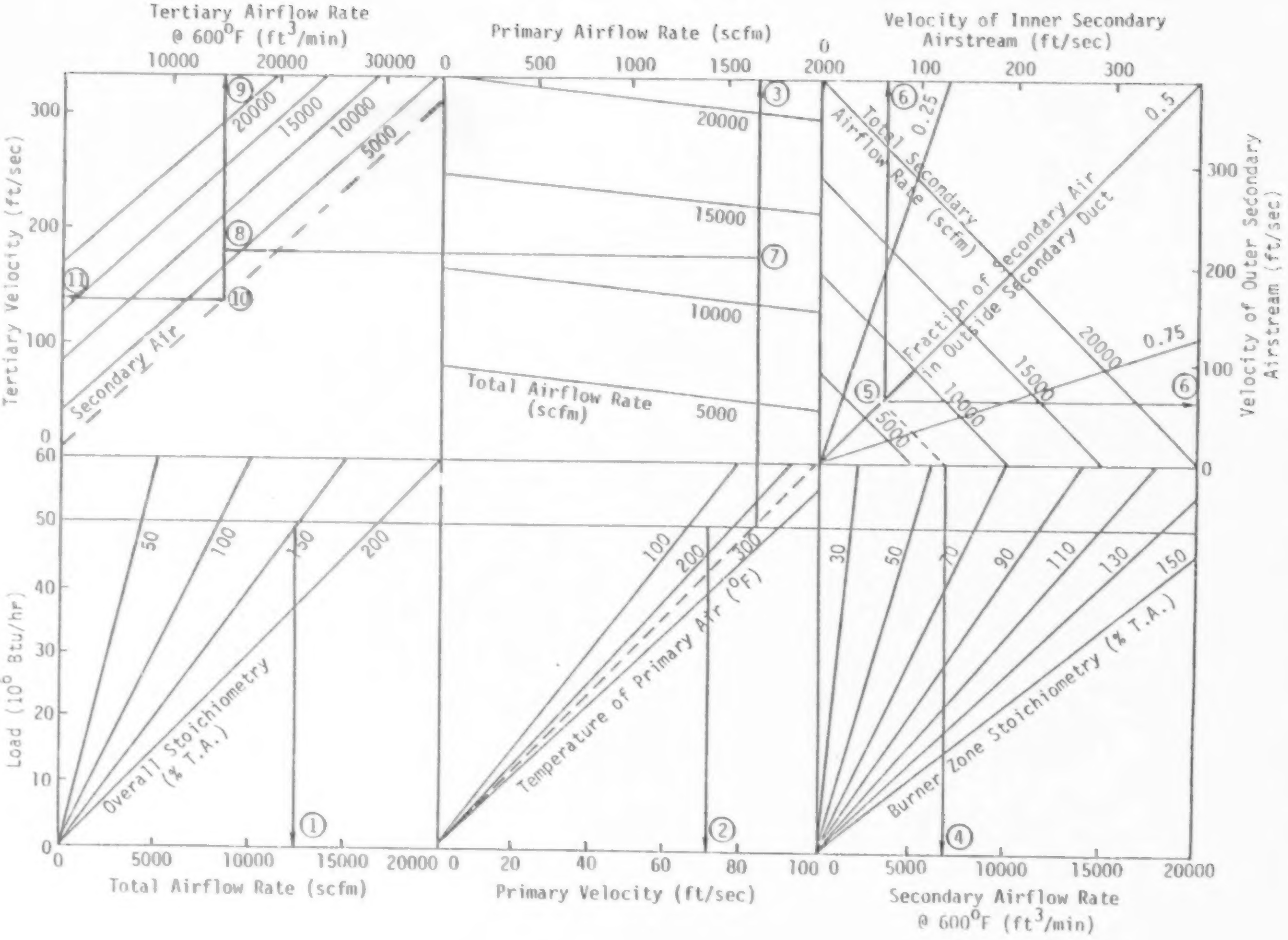


Figure A-4. Operating Map for 50 MBtu/hr Dual Secondary Throat Burner with Utah Coal, Primary Stoichiometry 20% T.A. and Combustion Air Temperature 600°F.

Due to the unique relationship between load and primary air flow rate (with the assumed primary stoichiometry of 20 percent T.A.) then the intersection of the load line with the dashed line in the center lower chart defines primary air flow rate ③ of 1,670 SCFM. To determine secondary flow parameters the upper right chart is entered by simply extending the vertical line which intersected the load and burner zone stoichiometry lines. The value of the total secondary air flow rate ⑤ can then be determined by interpolation between the two bracketing parallel lines. The value of total secondary air flow rate is 7,000 SCFM. Also, at the intersection of the dashed line which represents the total secondary air flow rate and the diagonal line indicating the assumed split between the inner and outer secondary channels of 50 percent, the velocities of the inner and outer secondary flow rates ⑥ are defined as 70 ft/sec and 70 ft/sec respectively. The remaining variables to be defined are those associated with the tertiary air. The tertiary air flow rate and velocity are defined by first locating in the upper middle chart the intersection of the vertical line defining the primary air flow rate and the value previously determined for total air flow rate (12,500 SCFM) ⑦ and then proceeding horizontally and entering the upper left chart until the line intersects the previously determined value of the total secondary air flow rate (7,000 SCFM) at ⑧. The tertiary air flow rate is then determined directly from the upper scale at ⑨ and is 15,000 ft^3/min. Finally, the tertiary air velocity is defined by first extending the vertical line which defined the tertiary air flow rate until it intersects the dashed diagonal line at ⑩ and reading from the ordinate at ⑪ the tertiary air flow rate is 140 ft/sec.

Use of this chart allows the engineer to quickly determine the impacts of varying specific burner flow parameters on the remaining variables. For example, referring to Figure A-4, it can be seen that as the burner zone stoichiometry becomes more fuel-rich, the velocities in the secondary air channels are reduced. However, the velocity in one secondary air channel can be maintained constant by diverting air from the other channel. Although Figure A-4 provides values for secondary velocities, the calculations only consider the axial components. As shown in Figure A-5, the 50 x 10^6 Btu/hr burner allows a wide variation in the distibution of both axial and tangential velocity in the secondary airstreams via the variable independent swirl in each of the two channels. It is important to note from the information presented in Figure A-4 that with fixed geometry, it is not possible to vary a single parameter independent of all others. As a consequence,

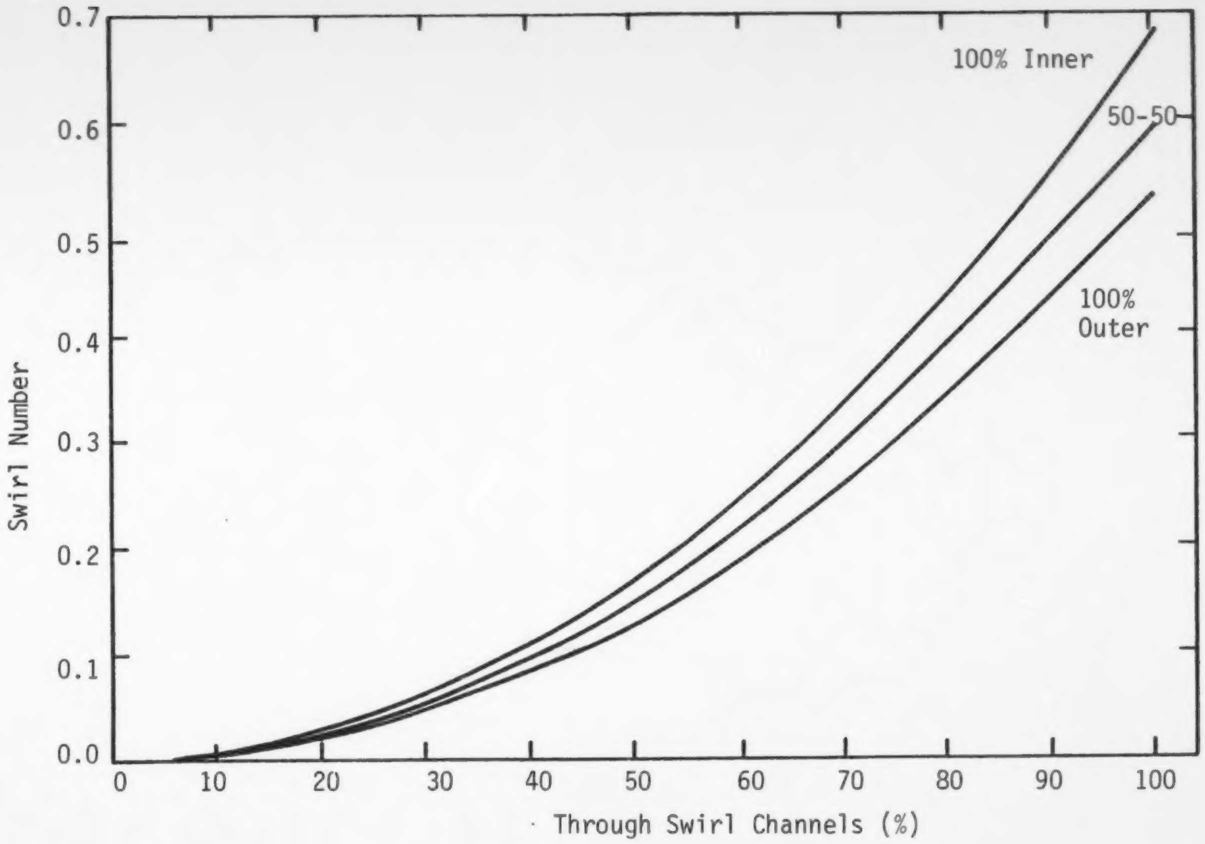


Figure A-5. Swirl Number for Dual Secondary Throat Burner (50 MBtu/hr).

optimization of performance and emissions will be specific to this test burner; however, the trends should be valid for all DMB designs.

The ability to vary secondary swirl over a wide range provides for considerable control over flame shape and the three basic flow patterns shown in Figure A-5a were observed in both cold and hot-flow testing. These were:

A-5

- Type A with all axial flow except for an annular toroidal recirculation zone, which provided for flame stability.
- Type B with an axial toroidal recirculation zone.
- Type C in which the flow attached to the combustion chamber wall produced a wall jet.

The transition from A through C occurred as the swirl level in the other channel increased. Low NO_x operation with this burner was generally achieved with a relatively long diffusion type flame, as shown in the sketch on Figure A-6b. In the combustor configuration employed in the tests with the 50 MBtu/hr burner, wherein the back wall was 16 feet from the firing wall, under some conditions the flame would impinge on the back wall. This condition accompanied increased build-up of residue in the bottom of the furnace, as well as increased opacity of the flue gases exiting the unit. Therefore, this burner was generally tested only at conditions where the wall impingement did not result in significant amounts of black smoke.

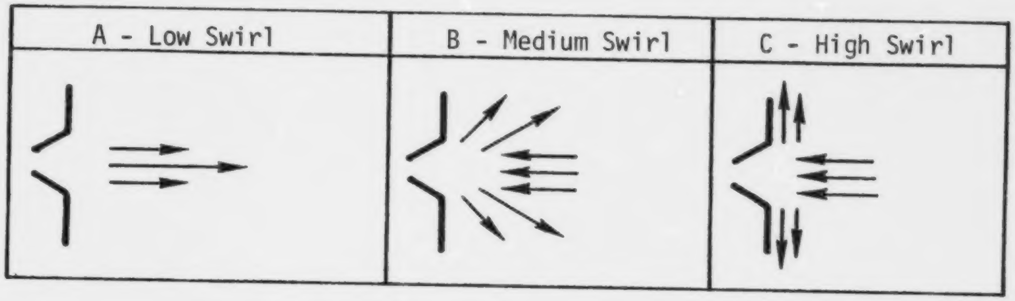
A.2 Single Secondary Burner

The single secondary burner design shown in Figure A-7 incorporates the following features.

A 6

- Three independently-controlled airstreams to divide the total combustion air into:
  - the primary air carrying the coal,
  - a concentric secondary airstream,
  - four outboard staged airstreams.
- A central fuel injector with a retractable oil gun on the burner axis.
- Swirl vanes at the exit of the fuel nozzle to control the rate of mixing of the fuel into the secondary airstream.
- The swirl level of the secondary airstream was controlled by varying the adjustable swirl vanes. Thus, the total swirl level can be varied from zero to a maximum dictated by the angle of the adjustable swirl vanes.

A. Flame Types


A - Low Swirl
B - Medium Swirl
C - High Swirl

B. Low NO_x Flame Shape - High Fineness Ratio Diffusion Flame

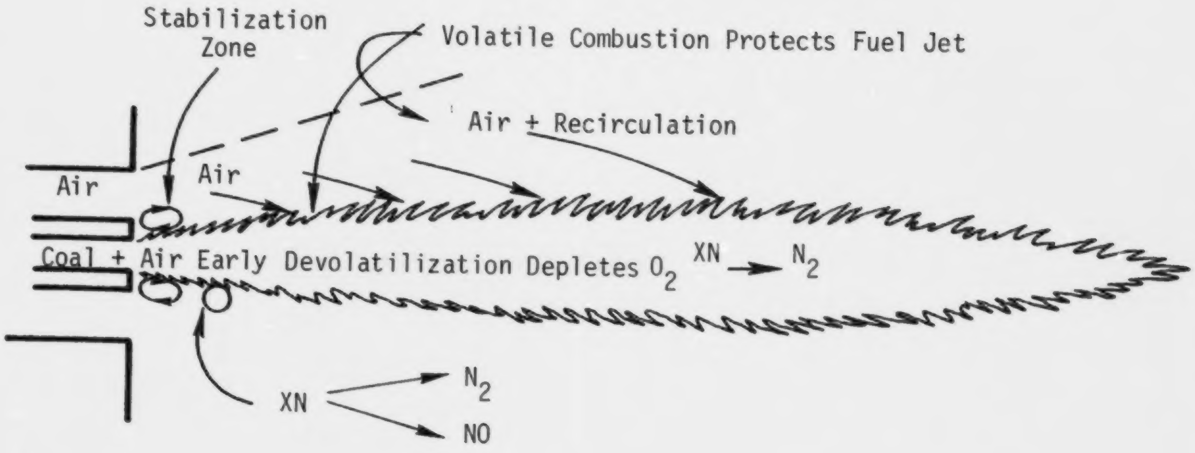

Stabilization Zone
Volatile Combustion Protects Fuel Jet
Air + Recirculation
Air
Air
Coal + Air Early Devolatilization Depletes O_2
$XN \rightarrow N_2$
XN
N_2
NO

Figure A-6. Flame Characteristics with Divided Throat Burner.

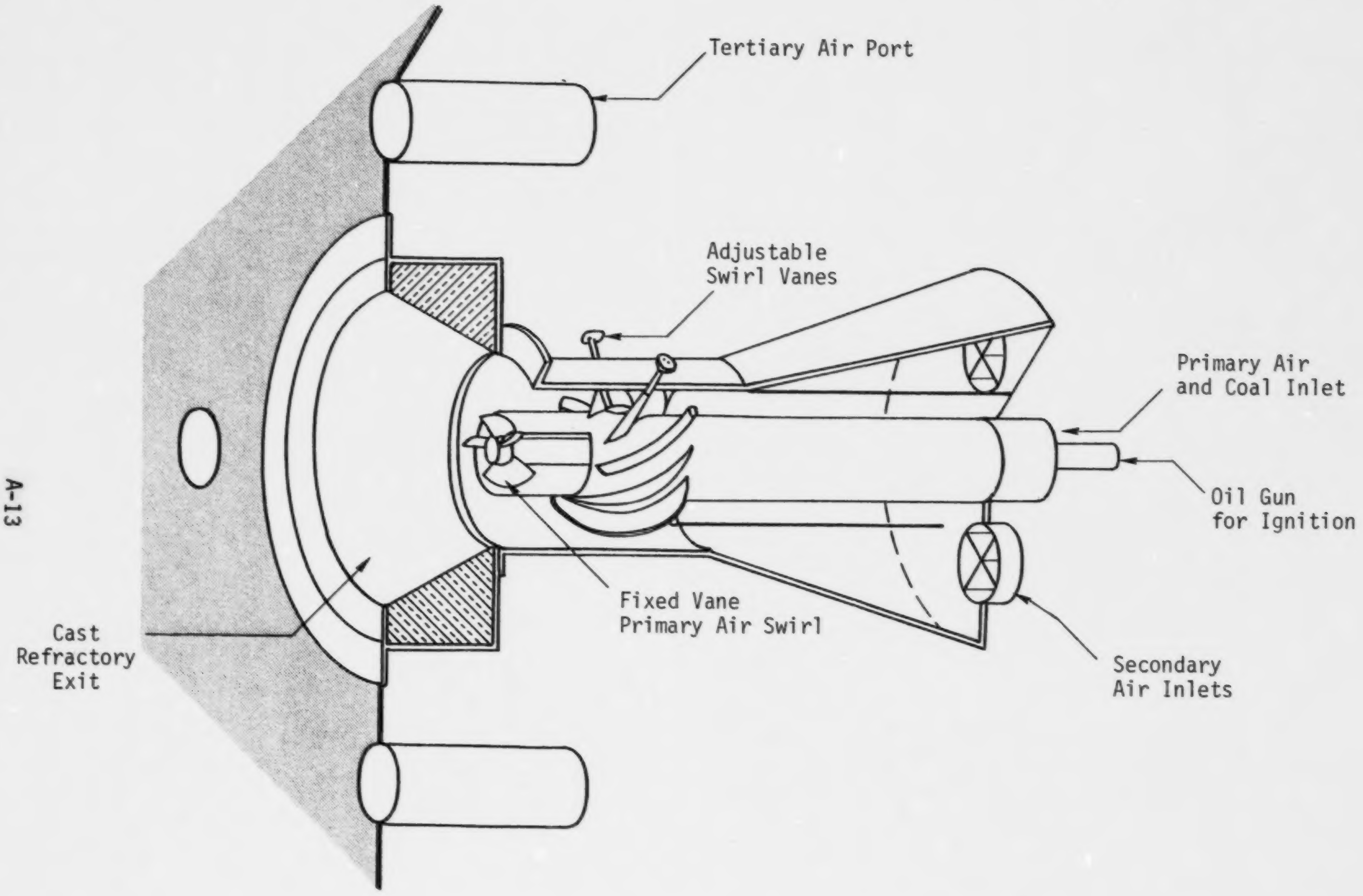

Tertiary Air Port
Adjustable
Swirl Vanes
Primary Air
and Coal Inlet
Oil Gun
for Ignition
Fixed Vane
Primary Air Swirl
Secondary
Air Inlets
Cast
Refractory
Exit

Figure A-7. Single Secondary Burner Design.

- Interchangeable ceramic burner exits.

The single secondary burner shown in Figure A-7 is a simple design incorporating practical features to provide ease of manufacture and operation, and to allow for the most straightforward interpretation of results. In order to develop scale criteria, the burners were designed at 10, 12.5, 50 and 100 x 10^6 Btu/hr firing rates. The principle of geometric similarity with constant velocity was used for scaling. The four burner scales each employed similar distributions of tangential velocity and similar locations of tertiary air inlets, although the 10 x 10^6 Btu/hr burner utilized the swirl method employed for the dual secondary burner. Further, the 10 x 10^6 Btu/hr burner did not employ a swirl vane in the coal nozzle. A summary of the specific design dimensions for each burner is provided in Table A-2.

Figure A-8 shows an operating map for the 50 Btu/hr version of the single secondary burner design. This figure is similar to that of the previous burner; however, the single secondary passage results in less complexity. The general flow characteristics and limitations are similar. In the example shown in Figure A-8 the assumed conditions are:

(1) Load = 50 x 10^6 Btu/hr

(2) Overall stoichiometry = 125% T.A.

(3) Primary Air Temperature = 150 F

(4) Burner zone stoichiometry = 70% T.A.

The horizontal line drawn at 50 x 10^6 Btu/hr intersects the other above parameters at ①, ③, and ④ allowing determination of the total air flow rate at ② of 10,600 SCFM, the primary air velocity at ⑤ of 70 ft/sec and the secondary air velocity at ⑥ of 88 ft/sec. The remaining parameters are then easily determined; for example, the Tertiary Air Velocity is determined by extending the horizontal line intersecting the Secondary Air Velocity with the previously determined Total Air flow rate (10,600 SCFM) until it intersects the previously determined primary air velocity inicated by the diagonal lines at ⑧. Note that the Tertiary Air Velocity can then be directly determined at ⑨ as 50 ft/sec.

A graph of the swirl index versus primary zone stoichiometry for two swirl angles (0 and 45 degrees) are shown in Figure A-9. As with the dual secondary throat, this flexibility allowed the three basic flow patterns previously described to occur. With the added flexibility of spreading the coal into the secondary air, however, a much shorter flame was achieved while maintaining the low NO_x

Table A-2. Design Information For Single Secondary Burners

Design Variable	Burner Capacity (10^6 Btu/Hr)			
	10	12.5	50	100
Fuel Injector				
Diameter (in)	5	5.5	10.5	15.75
Area (in^2)	19.63	24	86	195
Swirl Angle (Degrees)	Variable	45	45	195
Setback (in)	0	0	0	0
Secondary Air Channel				
Outer Diameter (in)	9.5	10.5	20	28
Annulus Thickness (in)	4.5	2.5	4.75	6.1
Area (in^2)	51.25	63	228	421
Swirl Vanes (degrees)	Variable	60	60	60
Setback (in)	0	0	0	0
Tertiary Ducts				
Distance from Burner C.L. (in)	—	22	44	62
Spacing Around Burner (degrees)	22.5	90	90	90
Number of Ports	16	4	4	4
Injection Angle (degrees)	0	0	0	0
Axial Position (in)	0	0	0	0
Diameter (in)	4.5	6	12	16
Total Area (in^2)	254.5	113	452	804
Throat and Exit				
Throat Diameter (in)	9.5	10.5	20	28
Throat Area (in^2)	71	87	314	616
Half Angle of Exit (degrees)	25	25	25	25
Length of Exit (in)	9	9 (18.5)*	18.5	27

*Alternate Exit Length to Match 50 X 10^6 Btu/Hr Burner Residence Time

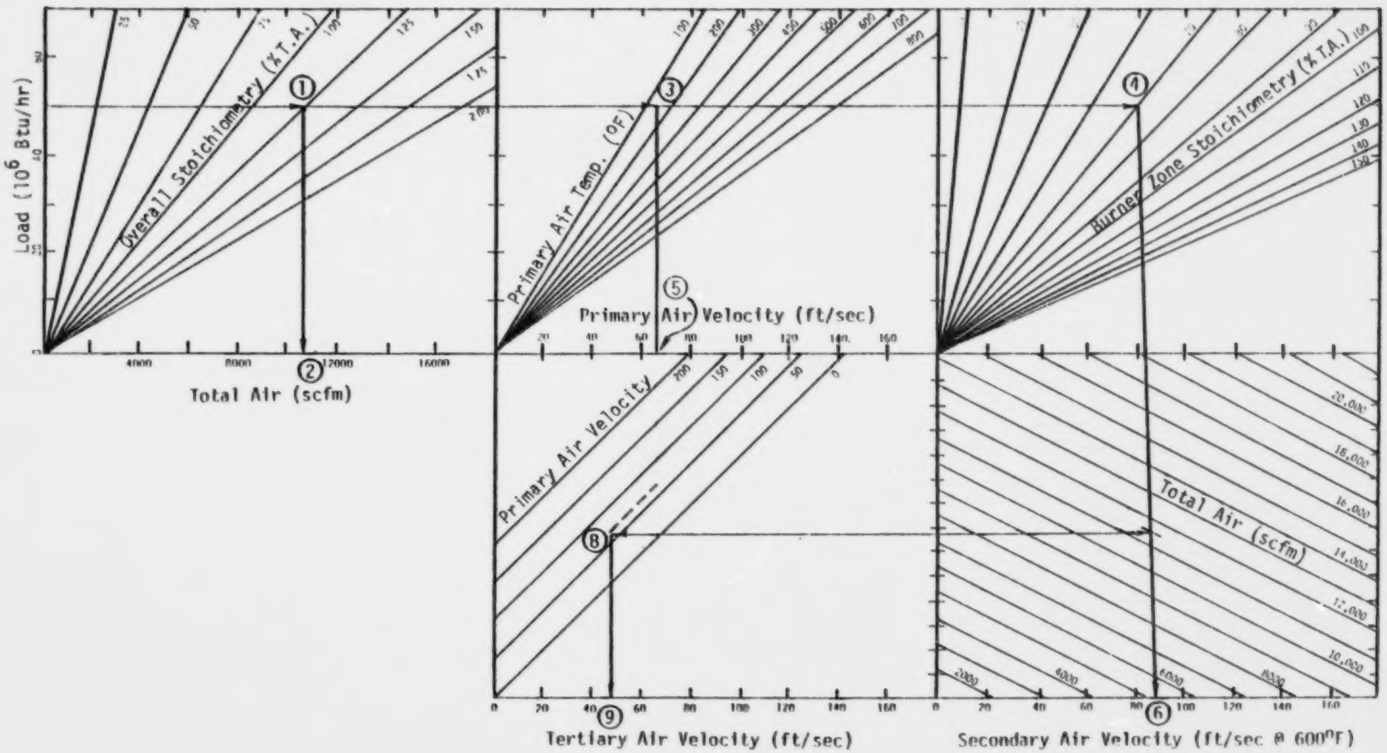


Figure A-8. Operating Map for Burner Air Velocities for Various Loads and Overall Stoichiometry-50 MBtu/hr Single Secondary Burner Design.

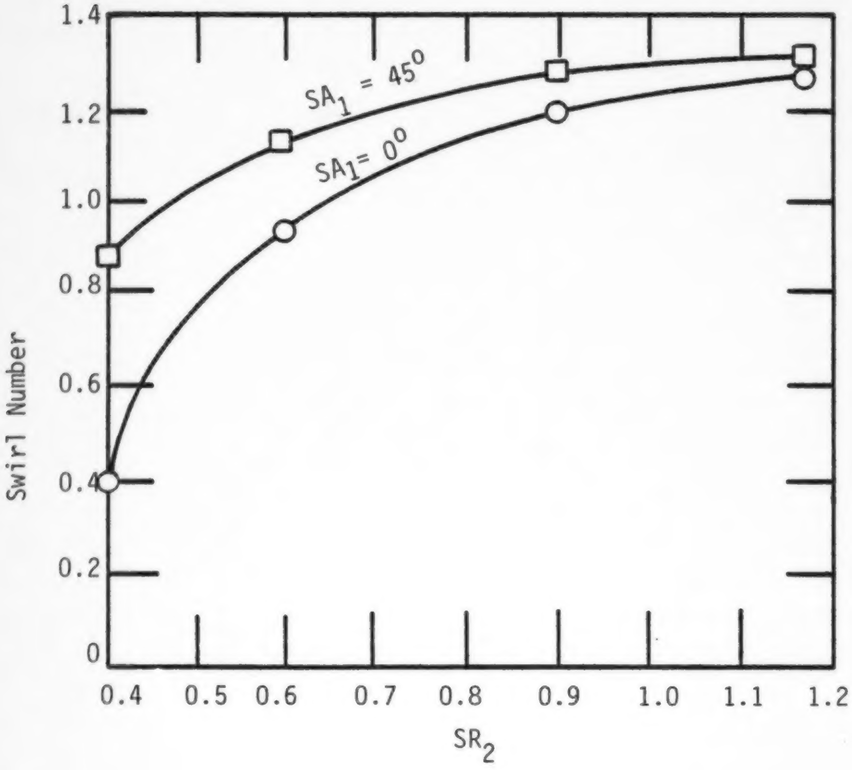


Figure A-9. Swirl Characteristics for the Single Secondary Burner Design.

characteristics. A sketch of the flame shape is given in Figure A-10. It should be noted that the shorter flame is compatible with most older boiler designs. The importance of this result is that low NO_x DMB designs can accommodate a broad range of boiler sizes since the flame shape/length can be varied from long diffusion-type flames compatible with deep furnaces including opposed-fired boilers to much smaller single wall-fired boilers.

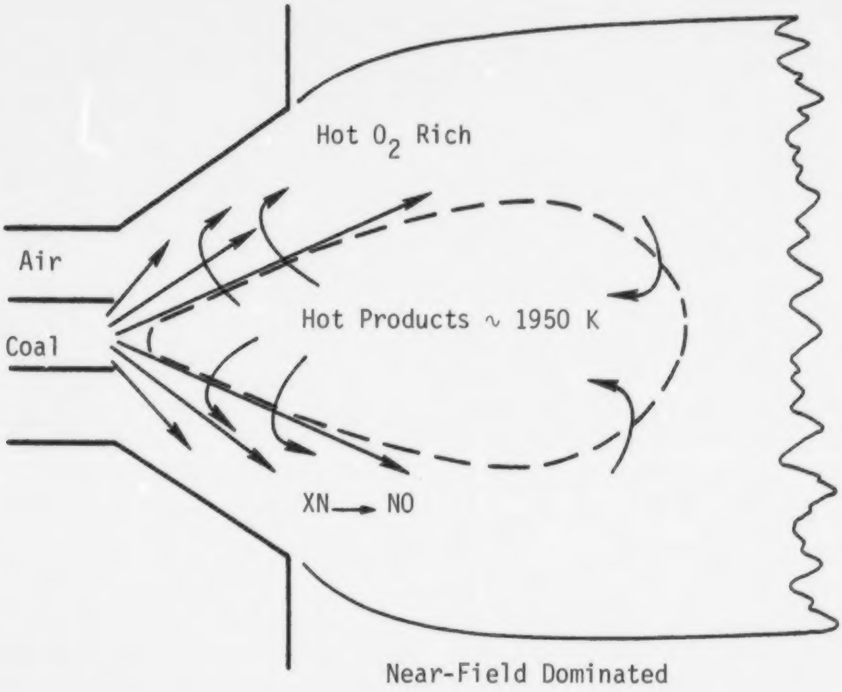

Hot O_2 Rich
Air
Coal
Hot Products ∿ 1950 K
XN → NO
Near-Field Dominated

Figure A-10. Typical Flame Shape for Single Secondary Burner.

APPENDIX B--RESEARCH FURNACE DESIGNS

B.1 Large Watertube Simulator (50 to 120 x 10^6Btu/Hr)

Full-scale simulation of the geometry and thermal environment of a utility watertube boiler is provided by the Large Watertube Simulator (LWS). The LWS facility layout with support systems is illustrated in Figure B-1. The LWS has been used at firing rates ranging from 50 to 100 x 10^6 Btu/hr and has multi-fuel capability. The combustor is 54 feet tall, 28 feet deep and 16 feet wide. The LWS incorporates a movable back wall to provide for a fourfold variation of the furnace volume as indicated in Figure B-2, and a range of firebox residence times can be obtained that are comparable with field utility boilers. The construction and variable geometry also provide the flexibility needed for investigating multiple burner arrays.

The outer shell and the movable back wall of the combustor are fabricated from large sections of preformed, corrugated mill steel sheet. The structure is stabilized by welded buckstays which are also formed from steel sheet. The shell is completely unrestrained in regard to thermal growth in the vertical direction, and corrugations provide for growth in the horizontal direction. The walls of the combustion chamber are cooled by water sprays supplied from manifolds incorporated into the main structural frame, spraying on the outside of the combustion chamber walls. Each spray head is controlled by a solenoid valve which is operated in a pulsed duration modulation mode. The spray controls are divided to operate sprays in several zones, thus allowing flexibility of wall temperature distribution control.

Airflow to the combustor is supplied by blower driven with an internal combustion engine. Because the experimental coal burners require multiple locations for air injection, a manifolding system is provided which allows the combustion airflow to be controlled and monitored in many individual streams.

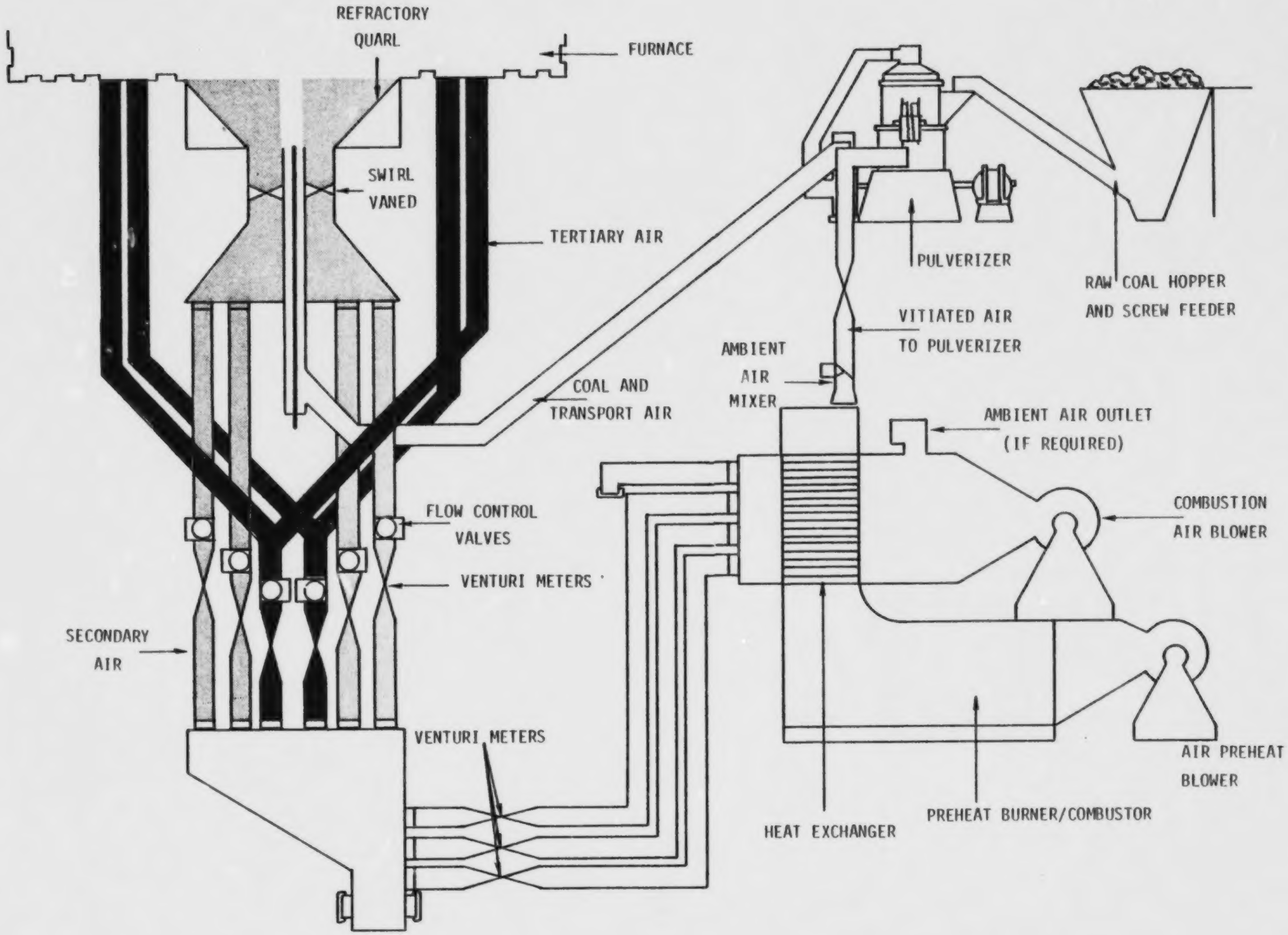

REFRACTORY QUARL
FURNACE
SWIRL VANED
TERTIARY AIR
PULVERIZER
RAW COAL HOPPER AND SCREW FEEDER
VITIATED AIR TO PULVERIZER
AMBIENT AIR MIXER
COAL AND TRANSPORT AIR
AMBIENT AIR OUTLET (IF REQUIRED)
COMBUSTION AIR BLOWER
FLOW CONTROL VALVES
VENTURI METERS
SECONDARY AIR
VENTURI METERS
AIR PREHEAT BLOWER
HEAT EXCHANGER
PREHEAT BURNER/COMBUSTOR

Figure B-1. Large Watertube Simulator (LWS) Test Facility.

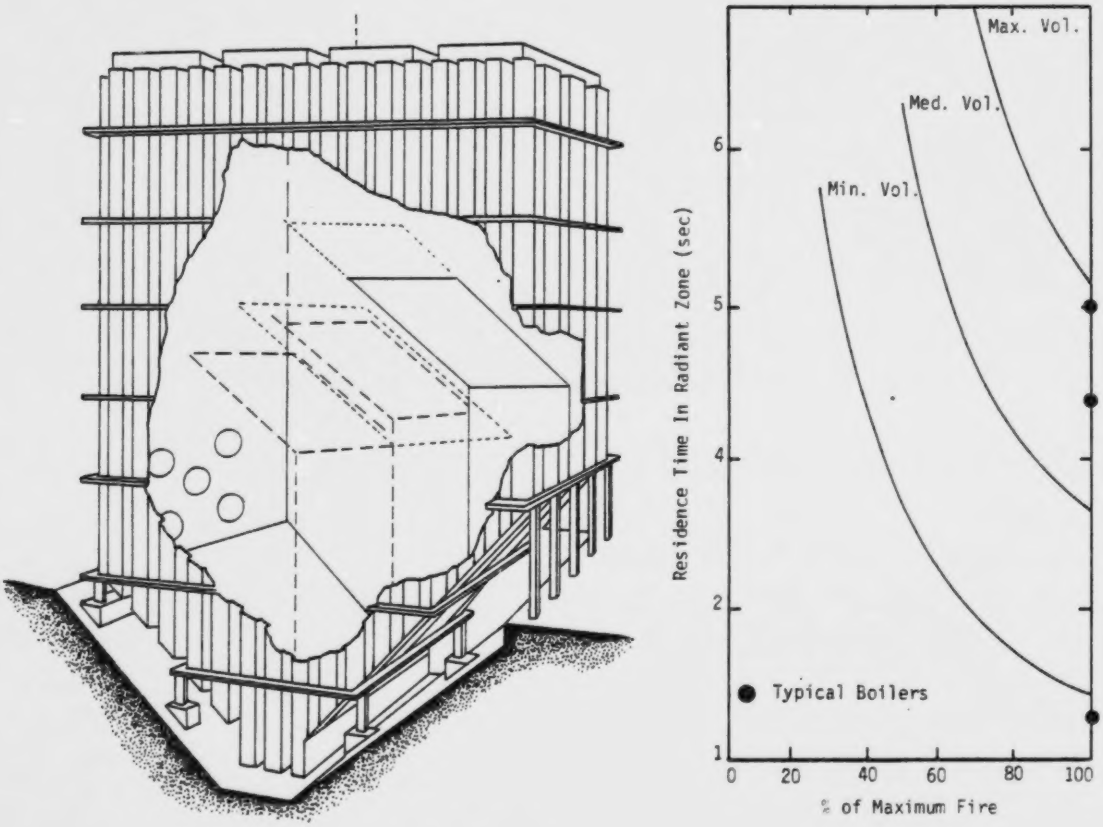

Max. Vol.
Med. Vol.
Min. Vol.
Residence Time In Radiant Zone (sec)
6
5
4
2
1
● Typical Boilers
0
20
40
60
80
100
% of Maximum Fire

Figure B-2. Variable Geometry LWS Combustor.

A schematic representation of the separate burner air streams which can be monitored is presented in Figure B-3. The total airflow is made up of:

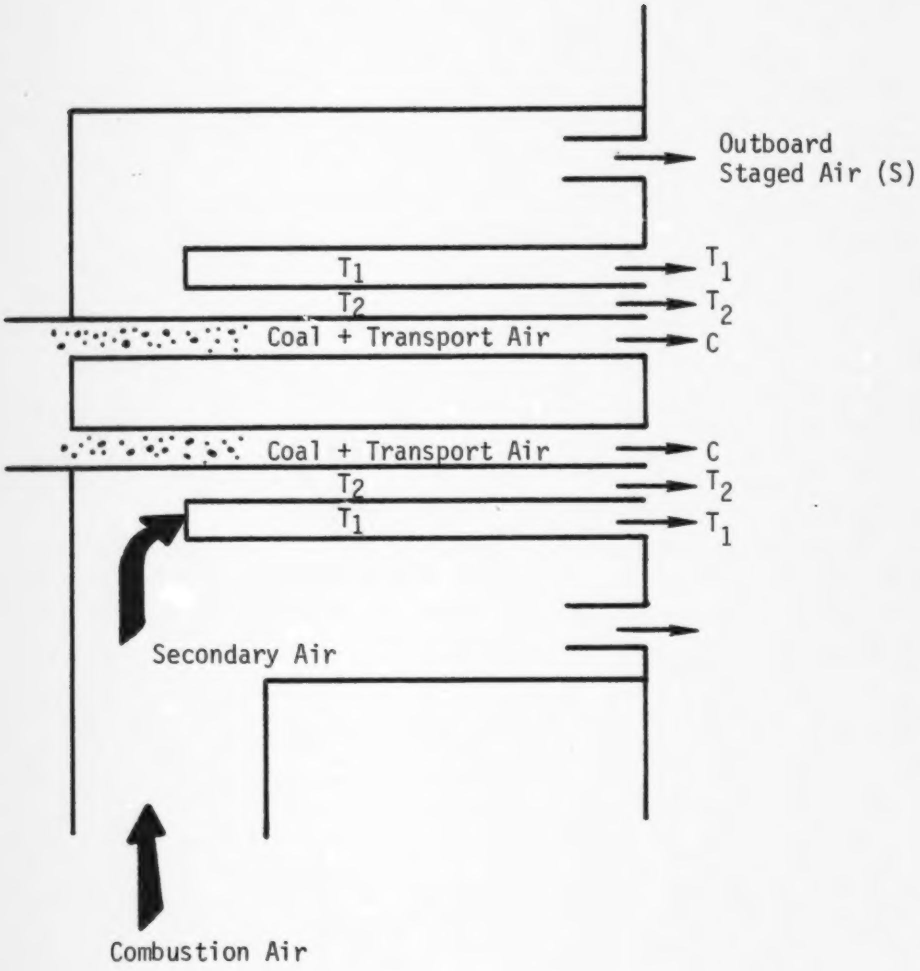
- Primary air which is that required to transport the coal from the pulverizer to the burner;
- Secondary air which is supplied through an annular throat surrounding the coal injector. This stream can be split into an inner and outer flow to allow for controlled mixing in the early part of the flame; and
- Outboard staged air which provides the burnout air and oxidizing envelope for the low emission flame.

The airflow in the various streams is monitored during the test program by individual venturis. For multiple or commercial burners, the airflow is monitored by calibrating the airflow as a function of pressure within the windbox. During the system checkout, pitot probe traverses were made in the various ducts on several dimetral planes, both as a check against the airflow calibration and to ensure uniform flow within each duct. Preheated combustion air is supplied from a separately fired preheater. The combustion air supply can be preheated to a maximum temperature of 700°F independently from the firing rate of the combustor. Solids are processed by a commercial Raymond CE pulverizer which has a capacity of about six tons/hr. The fuel is transported pneumatically from the pulverizer to the combustor.

B.2 Small Watertube Simulator Facility

The tests at 12×10^6 Btu/hr heat input with the low NO_x burner were carried out in the thermal environment typical of water wall boilers in a small watertube simulator (SWS). The geometry of the simulator is trapezoidal which is similar to the most popular design of small watertube boilers. The unit is schematically illustrated in Figure B-4, along with a layout of the support systems. The SWS can be fired at rates ranging from 4 to 16×10^6 Btu/hr. Other features of the SWS include:

- a ceramic front wall which can be moved to vary the furnace depth up to its maximum of 11 feet;


Outboard
Staged Air (S)
T_1
T_2
Coal + Transport Air
C
Coal + Transport Air
C
T_2
T_1
Secondary Air
Combustion Air

Total Airflow = Primary Air + Secondary Air + Outboard Staged Air
C $(T_1 + T_2)$ S

Figure B-3. Airflow Distribution.

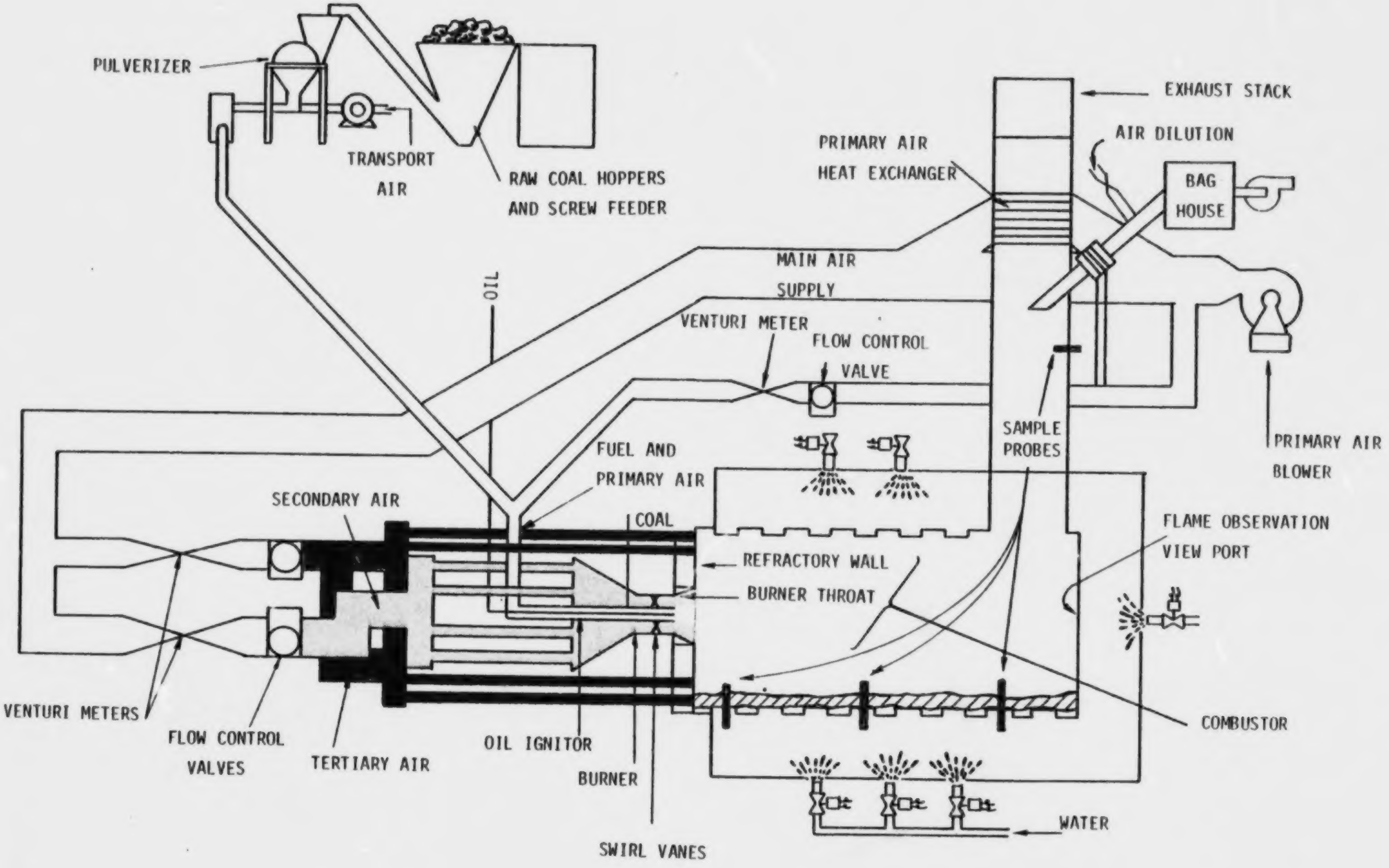

PULVERIZER
TRANSPORT AIR
RAW COAL HOPPERS AND SCREW FEEDER
OIL
EXHAUST STACK
AIR DILUTION
PRIMARY AIR HEAT EXCHANGER
BAG HOUSE
MAIN AIR SUPPLY
VENTURI METER
FLOW CONTROL VALVE
SAMPLE PROBES
PRIMARY AIR BLOWER
FUEL AND PRIMARY AIR
SECONDARY AIR
COAL
REFRACTORY WALL
BURNER THROAT
FLAME OBSERVATION VIEW PORT
COMBUSTOR
VENTURI METERS
FLOW CONTROL VALVES
TERTIARY AIR
OIL IGNITOR
BURNER
SWIRL VANES
WATER

Figure B-4. Small Watertube Simulator (SWS) Test Facility.

- sheet steel side and back walls which can be cooled by a pulse water spray system;
- a stack heat exchanger that uses the high temperature combustion products to preheat the incoming combustion air up to 600°F; and
- multi-fuel capability.

The coal is pulverized by a high-speed hammer mill and pneumatically transported cold to the burner. The fineness of the coal is controlled by screens. After pulverization, the fuel is pneumatically conveyed directly from the pulverizer to the combustor at a rate of up to 1000 lb/hr. Individual control and measurement are provided for each of the numerous air and fuel streams necessary to the staged burner concept. The airflow to various burner input streams is controlled in the same manner as described for the LWS.

B.3 Data Acquisition System

Data acquisition/reduction and control of burner/furnace operations are accomplished from the control panel illustrated in Figure B-5. The output from the various sensors and analyzers is sent to a data logger in series with an HP Model 9825A desk computer. A multipen recorder is used as a continuous signal monitor and recorder to the gas analyzers' output. Typical output from the desk computer is shown in Figure B-6. The contents of the output, as numerically labeled in Figure B-6, are described below:

- Input of fuel properties - required if not a routine fuel;
- Date;
- Combustor name for program code selection;
- Fuel combustion values;
- Burner systems description and dimensions;
- Range settings of analyzers;
- Complete temperature scan - optional;
- Warning messages;

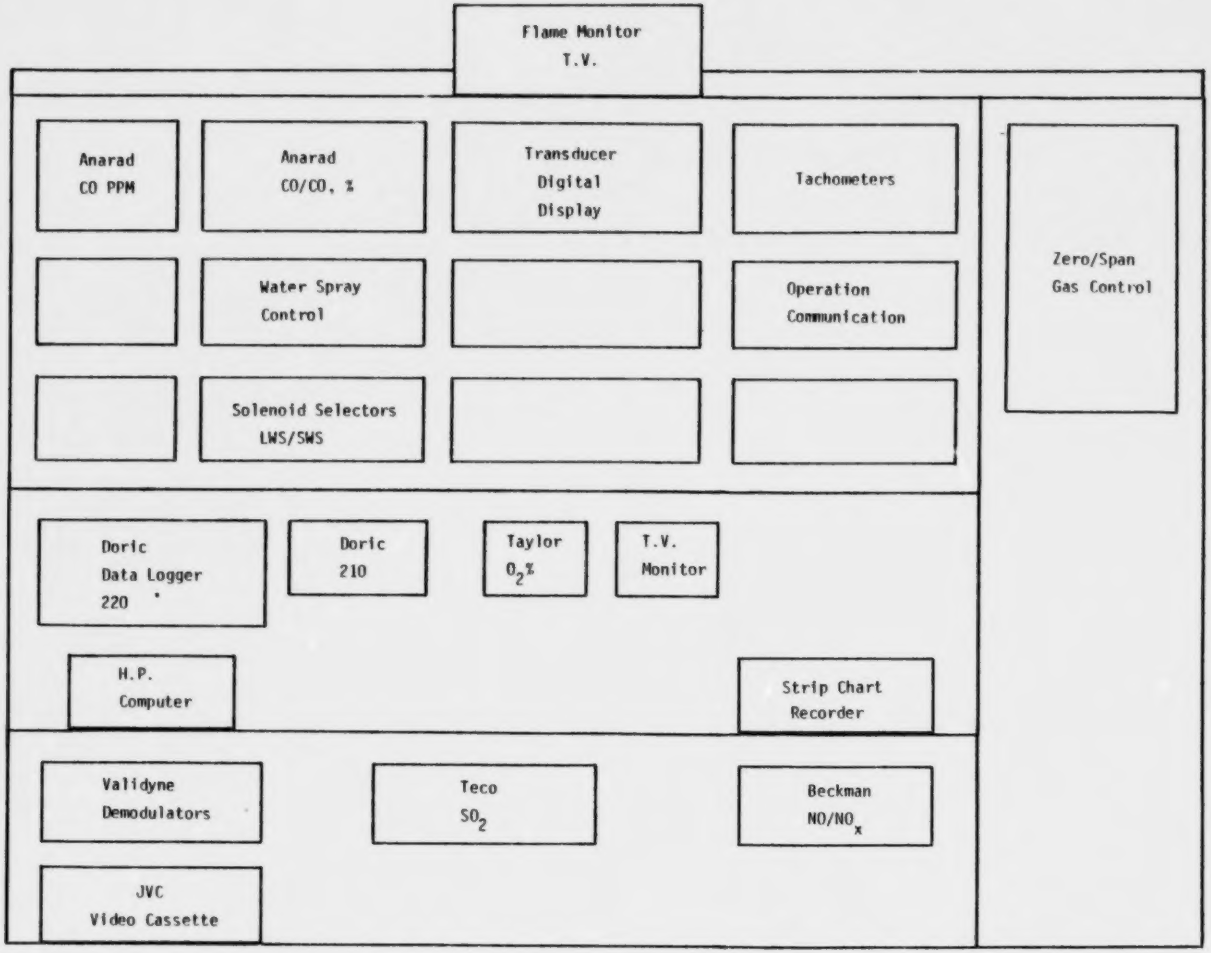
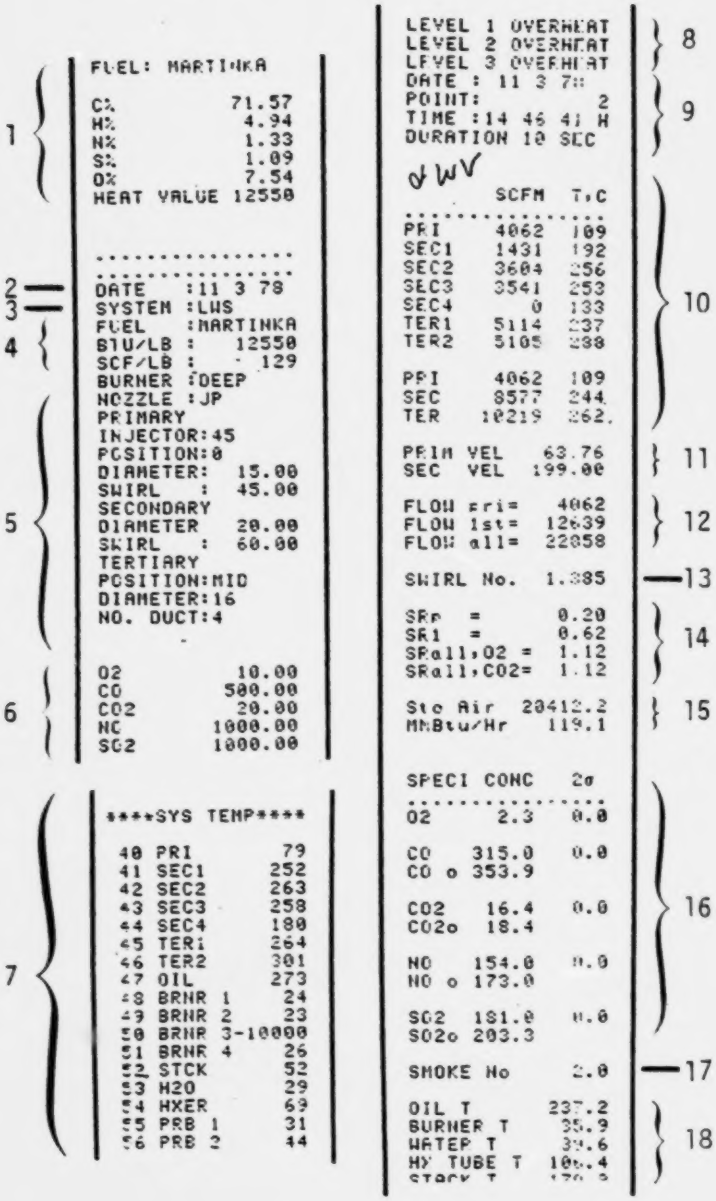

Flame Monitor
T.V.
Anarad
CO PPM
Anarad
CO/CO, %
Transducer
Digital
Display
Tachometers
Zero/Span
Gas Control
Water Spray
Control
Operation
Communication
Solenoid Selectors
LWS/SWS
Doric
Data Logger
220
Doric
210
Taylor
O_2%
T.V.
Monitor
H.P.
Computer
Strip Chart
Recorder
Validyne
Demodulators
Teco
SO_2
Beckman
NO/NO_x
JVC
Video Cassette

Figure B-5. Control and Data Acquisition Layout.



```
FUEL: MARTINKA

C%           71.57
H%            4.94
N%            1.33
S%            1.09
O%            7.54
HEAT VALUE 12550


................
................
DATE   :11 3 78
SYSTEM :LWS
FUEL   :MARTINKA
BTU/LB :   12550
SCF/LB :   · 129
BURNER :DEEP
NOZZLE :JP
PRIMARY
INJECTOR:45
POSITION:0
DIAMETER:  15.00
SWIRL   :  45.00
SECONDARY
DIAMETER   20.00
SWIRL   :  60.00
TERTIARY
POSITION:MID
DIAMETER:16
NO. DUCT:4

O2           10.00
CO          500.00
CO2          20.00
NO         1000.00
SO2        1000.00
```

```
****SYS TEMP****

 40 PRI          79
 41 SEC1        252
 42 SEC2        263
 43 SEC3        258
 44 SEC4        180
 45 TER1        264
 46 TER2        301
 47 OIL         273
 48 BRNR 1       24
 49 BRNR 2       23
 50 BRNR 3-10000
 51 BRNR 4       26
 52 STCK         52
 53 H2O          29
 54 HXER         69
 55 PRB 1        31
 56 PRB 2        44
```

```
LEVEL 1 OVERHEAT
LEVEL 2 OVERHEAT
LEVEL 3 OVERHEAT
DATE : 11 3 78
POINT:         2
TIME :14 46 41 H
DURATION 10 SEC

         SCFM  T,C
................
PRI      4062  109
SEC1     1431  192
SEC2     3604  256
SEC3     3541  253
SEC4        0  133
TER1     5114  237
TER2     5105  238

PRI      4062  109
SEC      8577  244.
TER     10219  262.

PRIM VEL   63.76
SEC  VEL  199.00

FLOW pri=   4062
FLOW 1st=  12639
FLOW all=  22058

SWIRL No.  1.385

SRp   =     0.20
SR1   =     0.62
SRall,O2 =  1.12
SRall,CO2=  1.12

Stc Air  20412.2
MMBtu/Hr   119.1


SPECI CONC   2σ
................
O2      2.3   0.0

CO    315.0   0.0
CO o  353.9

CO2    16.4   0.0
CO2o   18.4

NO    154.0   0.0
NO o  173.0

SO2   181.0   0.0
SO2o  203.3

SMOKE No      2.0

OIL T       237.2
BURNER T     35.9
WATER T      39.6
HX TUBE T   106.4
STACK T
```

Figure B-6.. Desk Computer Output.

- Date, data point number, time and duration of sample scan - output;
- Individual and zone airflows - calculated from venturi calibration tables and measured temperatures;
- Calculated velocities using burner flows and dimensions;
- Total airflow (to each zone);
- Swirl number calculated from burner settings, dimensions and flows;
- Calculated stoichiometric ratios in the primary, primary-plus -secondary, and overall furnace based on oxygen and on carbon dioxide;
- Calculated stoichiometric airflow for the attendant firing rate;
- Sample gas (dried) composition as measured by the analyzers and the composition reduced to zero percent excess oxygen (subscript 0);
- Visual determination of smoke number - input;
- Temperatures desired for continuous ouput;

Various options are available for data acquisition which include sample data averaging over a specified time interval, data elimination by variation limits, etc.

B.4 General Output Measurements

Examples of output measurements being routinely measured during the program are:

- Gaseous pollutant and performance measurements. NO_x, SO_2, CO, unburned hydrocarbons (UHC), CO_2, O_2;
- Solid particle measurements. Solids extracted both from the fly ash and the bottom ash for analysis of N, C, S and ash composition;
- Flame characteristics such as flame shape, stability, in-flame and peripheral-flame performance, ignition and noise;

For each specific measurement, a determination has been made of the required frequency of these measurements consistent with technical requirements and cost-effectiveness. For this study, the frequency of the specific measurements is presented in Table B-1. In general, the continuous and/or intermittent measurements and recordings are made for all inputs, the usual gas species output and a comprehensive range of temperatures. This allows continuous definition of the overall low NO_x burner performance. In addition to the overall continuous measurements, intermittent measurements as indicated in Table B-1 are being performed for detailed characterization to evaluate the overall performances necessary for burner applications and the effects of low NO_x operation on other potential pollutants. The overall layout of the instrumentation for the various facilities is illustrated in Figure B-7 and are described below.

The exhaust gas composition is determined using the instrumentation listed in Table B-2. The flow schematic for the exhaust gas analyzers is presented in Figure B-8, and the sample conditioning system is shown in Figure B-9. Instrument zero and span may be accomplished by directing calibration gases to the instruments or to the entry of the sample train. The former provides for direct calibration of each instrument, whereas the latter is essentially a calibration of the entire sample train. Sample conditioning consists of water and particulate removal. Two water refrigerator traps are used to provide water removal as soon as possible from the sample and after compression by the sample pump. A cyclone and/or filters of various collection efficiencies and materials are used for particulate removal. The final filter provides for 99.99 percent retention of 0.6 micron or larger particles. Single orifice probes with a water cooling jacket are made of stainless steel. In the multiple probe array used in the LWS, the probes are manifolded externally so that any individual probe can be monitored. Teflon lines are used for sample and calibration gas transfer.

Gas samples from the combustors are withdrawn from the sample probe(s) and flow vertically downwards through an ice bath. The bath consists of a five-foot vertical length of Teflon tubing surrounded by ice water. This tubing terminates with a tangential throw tube into a glass water trap of

Table B-1. General Measurement Scheme

	Routine		Non Routine			
Gas	CO, CO_2, O_2, NO	SO_2	UHC, NO_x SO_x, HCN, NH_3	Complex Organics XS	—	
Particulate	None		C, H, N Ash	Sulfur Mass Size	Morphology Trace Elements, Large Organic Material	
Physical	Temperature		—	Velocity	—	
Variables	Many ⟶ Limited					

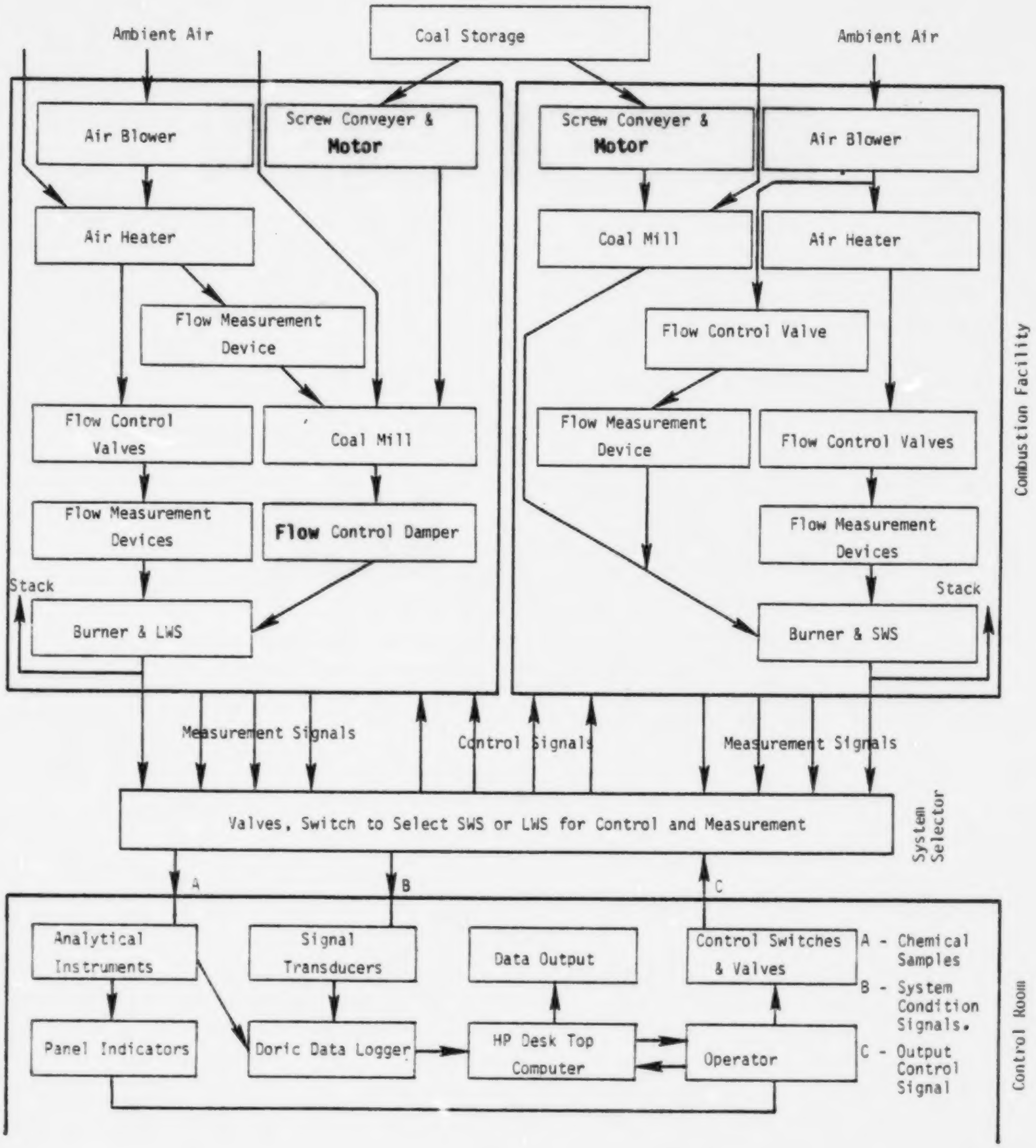

Coal Storage
Ambient Air
Ambient Air
Air Blower
Screw Conveyer & Motor
Screw Conveyer & Motor
Air Blower
Air Heater
Coal Mill
Air Heater
Flow Measurement Device
Flow Control Valve
Flow Control Valves
Coal Mill
Flow Measurement Device
Flow Control Valves
Flow Measurement Devices
Flow Control Damper
Flow Measurement Devices
Stack
Stack
Burner & LWS
Burner & SWS
Combustion Facility
Measurement Signals
Control Signals
Measurement Signals
Valves, Switch to Select SWS or LWS for Control and Measurement
System Selector
A
B
C
Analytical Instruments
Signal Transducers
Data Output
Control Switches & Valves
Panel Indicators
Doric Data Logger
HP Desk Top Computer
Operator
A - Chemical Samples
B - System Condition Signals.
C - Output Control Signal
Control Room

Figure B-7. Overall Scheme of Measurement Control of SWS/LWS Facilities.

Table B-2. Analytical Instruments for Combustion Gases

Species	Analytical Method/Principle	Accuracy	Range (Volumetric Concentrations)
CO_2	Non-Dispersive Infrared Analysis - differential spectral energy absorbed by sample gas relative to carrier gas is proportional to specific concentration	± 3%	0 - 40%
CO	Same	± 3%	0 - 12% 0 - 1,000 ppm
O_2	Paramagnetic Analysis - paramagnetic characteristics of a sample gas containing oxygen is porportional to the oxygen concentration	± 3%	0 - 100% 0 - 25% 0 - 10%
NO/NO_x	Chemiluminescent Analysis - intensity of spectral emission from chemiluminescence of excited state NO_2 (converted 0 - 1% from NO) is proportional to NO in sample gas	± 3%	0 - 10,000 ppm 0 - 2,500 ppm 0 - 1,000 ppm 0 - 250 ppm 0 - 100 ppm 0 - 10 ppm
SO_2	Pulsed Fluorescence - intensity of spectral emission from the fluorescence of excited state SO_2 is proportional to the SO_2 concentration	± 3%	0 - 5,000 ppm 0 - 1,000 ppm 0 - 500 ppm 0 - 100 ppm 0 - 50 ppm
HC	Flame Ionization Detector - electrical conductivity properties of a hydrogen flame doped with sample gas is proportional to the number of carbon atoms in the sample	± 3%	0 - 10,000 ppm 0 - 5,000 ppm 0 - 1,000 ppm 0 - 100 ppm

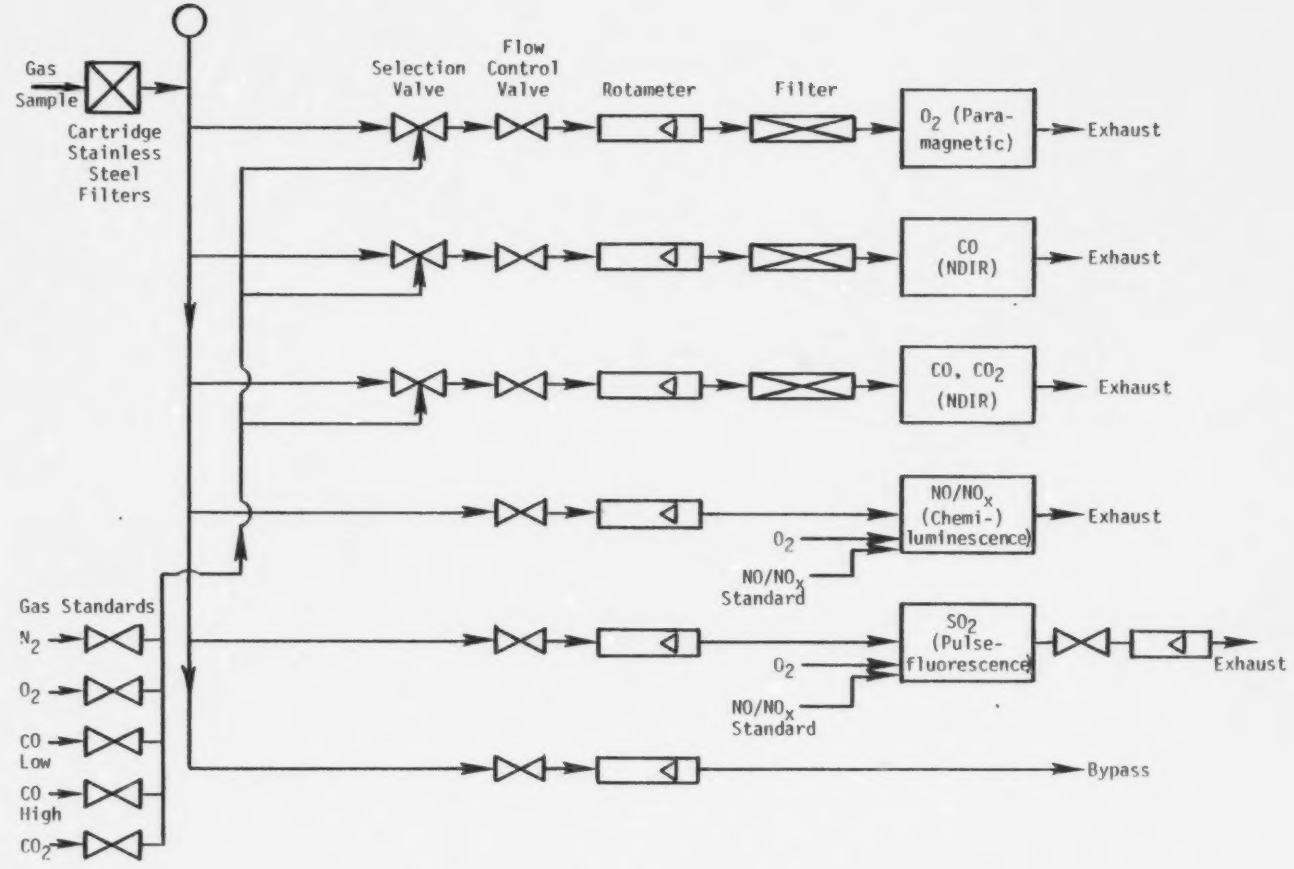

Gas
Sample
Cartridge
Stainless
Steel
Filters
Selection
Valve
Flow
Control
Valve
Rotameter
Filter
O_2 (Para-
magnetic)
Exhaust
CO
(NDIR)
Exhaust
CO, CO_2
(NDIR)
Exhaust
NO/NO_x
(Chemi-)
luminescence)
Exhaust
O_2
NO/NO_x
Standard
SO_2
(Pulse-
fluorescence)
Exhaust
O_2
NO/NO_x
Standard
Bypass
Gas Standards
N_2
O_2
CO
Low
CO
High
CO_2

Figure B-8. Flow Systems for Exhaust Gas Analyzers.

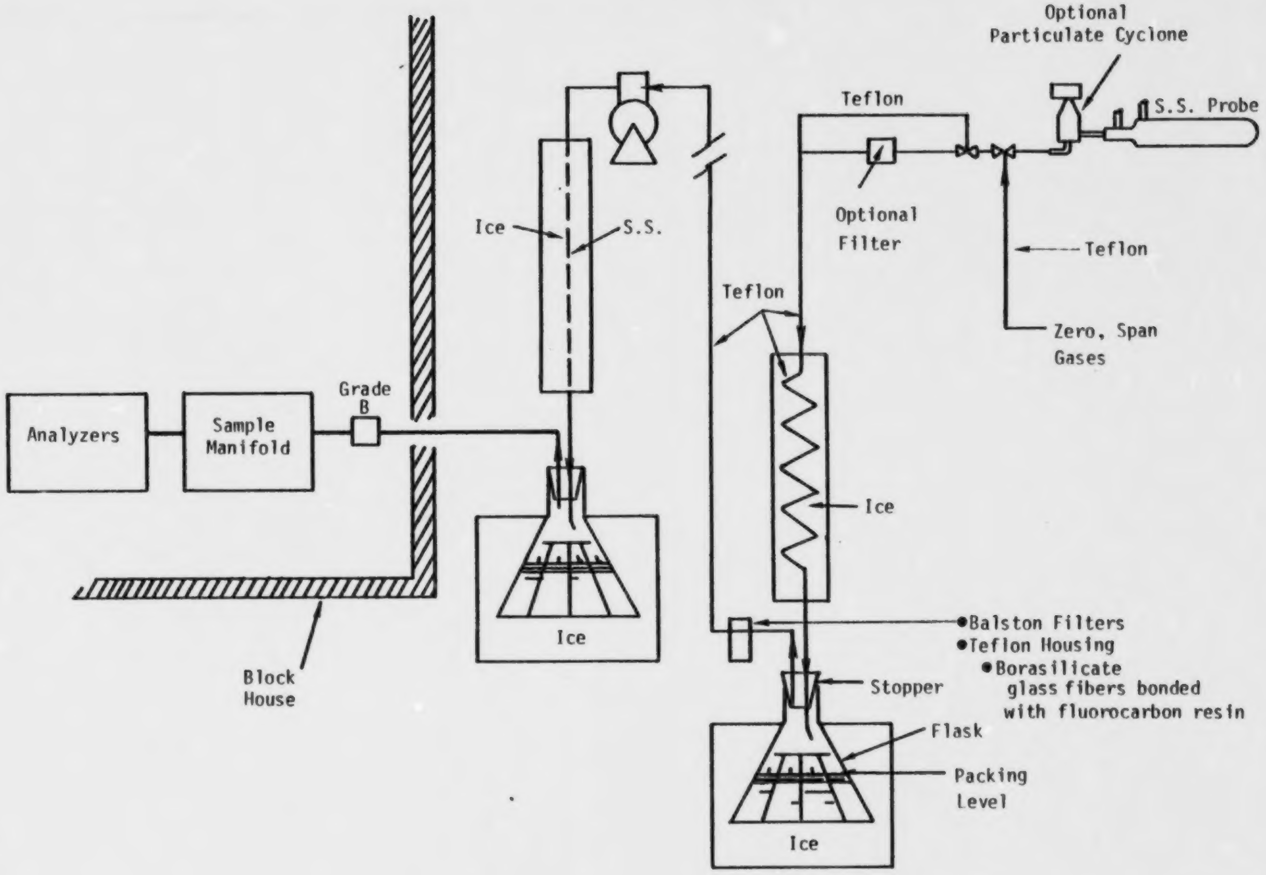

Optional
Particulate Cyclone
S.S. Probe
Teflon
Optional
Filter
Teflon
Zero, Span
Gases
Teflon
Ice
S.S.
Ice
Grade
B
Analyzers
Sample
Manifold
Ice
Block
House
●Balston Filters
●Teflon Housing
●Borasilicate
glass fibers bonded
with fluorocarbon resin
Stopper
Flask
Packing
Level
Ice

Figure B-9. Exhaust Gas Sample Conditioning System.

minimum volume (approximately 2.5 cubic inches or 40 cc) and residence time (approximately 0.05 sec). The gaseous output from the water trap is transferred in a Teflon tube to a dual head sample pump located atop the control house. After the pressurizing pump, the sample flow vertically downwards through a second ice bath which consists of a five-foot stainless steel (S.S.) tube surrounded by ice water. This S.S. tube terminates with a tangential throw tube into a S.S. water trap. The gaseous output from the second trap then flows through a filter to a sample manifold. Filters of various grades and materials, including spectro grade glass fiber (borosilicate glass fiber bonded with fluorocarbon resin) and Teflon, can be located anywhere along the sample train. Parallel samples from the sample manifold flow through regulation valves to the individual analyzers. Fine particulate filters are used upstream of the NDIR and paramagnetic instruments as well as all the sample and sample bypass flowmeters. The pressure and flow sensing meters supplied with individual instruments are maintained at the manufacturers' specifications.

Stable combustion species are analyzed with state-of-the-art commercial instruments. A list of the species that are routinely measured, the analytical technique and the accuracy of the instruments at well-maintained conditions are also contained in Table B-2. The continuous outputs are recorded on a multi-channel strip chart recorder. Other permanent recording and data reduction are performed with the on-line data logger and desk computer described earlier.

APPENDIX C--EFFECTS OF OPERATING VARIABLES AND FUEL COMPOSITION ON DMB PERFORMANCE

The development DMB test results presented in Section 3 of this report were conducted firing a Utah low-sulfur high-volatile bituminous coal at full load. This Appendix presents additional data obtained firing the development DMBs over a range of operating conditions with several fuels. The effects of operating conditions are discussed in the following section, and fuel effects are discussed in Section C.2. Section C.3 summarizes these results and shows the overall ranges of DMB operation achieved with each burner.

C.1 Operating Variables

For the purpose of this report, operating variables will be defined as those which are normally varied during typical burner operations in a boiler. They include:

- Overall excess air
- Burner heat release (load)
- Primary and secondary temperatures

In the burner development effort overall excess air was varied, keeping the burner operating conditions constant by adjusting the tertiary airflow rate. Although this results in changing both the overall excess air and tertiary air injection velocity, it was determined that this latter parameter did not influence the NO characteristics (see Section 3). Therefore, NO variations with excess air are likely to principally be the result of changes in conversion of the available XN species emerging from the burner zone, as well as any thermal NO contribution. The variations in CO, without varying the burner stability, should be related to changes in overall mixing uniformity.

An objective of the DMB development was to establish sufficient single and multiburner scaling criteria so that large and small capacity burners could be designed to provide identical fuel/air mixing characteristics. Therefore, for a fixed furnace configuration, NO_x would not be sensitive to burner capacity. Another objective was to establish the impact of the burner heat release rate on NO_x independent of burner scale by varying the furnace volume. These two physical changes provide information to assess the range of application of the DMB technology to existing and future boilers.

At this time the DMB has been evaluated in two furnaces: in the SWS at one burner size and in the LWS at (1) two burner sizes, and (2) a single and four-burner configuration.

In a boiler, primary and secondary air temperatures are determined by equipment design. The primary temperature is usually in the range of 150 to 160°F, and is determined by the pulverizer design and the need to have the coal sufficiently dry. The secondary air temperature is determined by the air heater design, as well as load and excess air, and is usually in the range of 500° to 600°F. During the development efforts air temperatures were maintained within these ranges and the effects of independently varying air temperatures were not examined.

C.1.1 Short Flame DMB Designs

Overall Excess Air

The effect of overall stoichiometry on NO characteristics for all single secondary short flame burners evaluated are shown in Figure C-1. F C-1 Note that the single and multiple burners behaved similarly in that increasing overall excess air results in increasing levels of NO. It is interesting that comparing the two single burners evaluated in the LWS, the 50 x 10^6 Btu/hr burner NO characteristics appear to be more sensitive to excess air than the 100 x 10^6 Btu/hr burners, suggesting that there might be a scale effect on the rate of mixing of the tertiary air with the burner air. The corresponding CO characteristics are shown in Figure C-2. F C-2 While the minimum levels achievable are dependent on the characteristics of the specific burner and can be changed by varying burner settings, the overall trends are identical. At high values of excess air the CO minimizes as would be expected. Decreasing excess air over a broad range in excess air does not influence CO until the overall mixing characteristics and flame characteristics of the burner become affected. At this point there is insufficient oxygen available to fully oxidize the carbon in the fuel and CO emissions increase rapidly. This same behavior has been observed with commercial burners operating in steam generators. However, the CO levels at high excess air are generally much lower. The higher CO levels in the EER research furnaces are believed to be a result of the cold furnace designs (low heat release per unit volume and per unit cooled surface area).

These results clearly show that the DMB should be operated at the lowest possible excess air commensurate with satisfactory CO and/or carbon burnout. This will minimize NO emissions and/or allow the burner to be operated at higher burner zone stoichiometries achieving the same NO level.

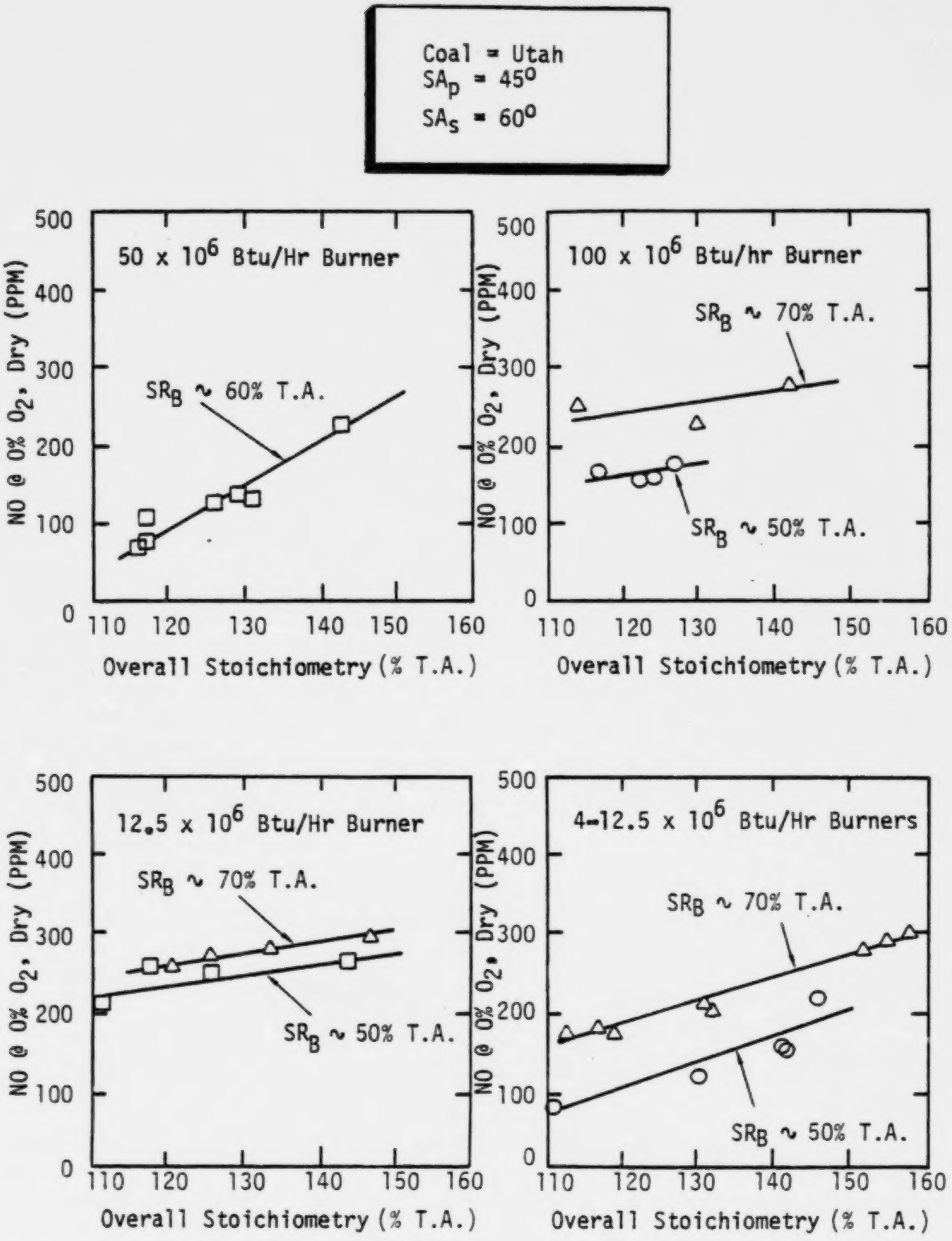


Figure C-1. Effect of Overall Stoichiometry on NO Characteristics for All Single Secondary - Short Flame Burners.

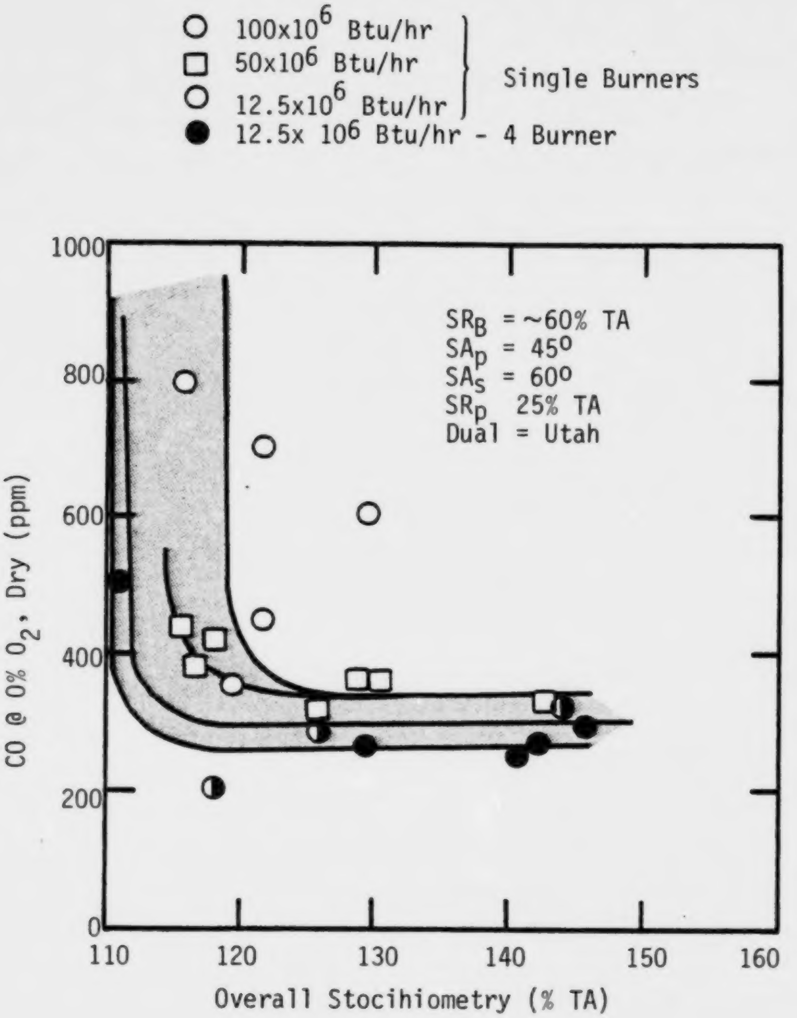


Figure C-2. CO Characteristics as a Function of Overall Stoichiometry for the Single Secondary - Short Flame Burners.

Burner Heat Release

Figure C-3 shows the relationship between NO emissions and burner heat release for the short flame DMB designs. These data were obtained over narrow ranges of operating conditions and are thus directly comparable. Although there is some scatter, the data for the three configurations tested in the LWS generally lie on a line of increasing NO emissions as heat release rate is increased. This increase in NO emissions rate as firing rate is increased has been observed in tests of many burners in research furnaces as well as field operating boilers. This phenomenon is due primarily to increased turbulent fuel/air mixing and a hotter furnace environment at higher firing rates. It should be noted that the LWS furnace has a cold thermal environment and that changes in firing rate over the factor of two ranges shown in Figure C-3 would not be expected to significantly affect NO generation.

Figure C-3 also shows data obtained in the SWS. Although these data do not cover a sufficient range of firing rates to identify a trend, they are considerably higher than the extrapolated trend of the LWS data. This is believed to be a consequence of the SWS furnace design (see Appendix B).

In summary, both burner firing rate and furnace design appear to influence NO emissions. However, it is clear that the burner designs tested allow satisfactory operation over a considerable range of firing rates.

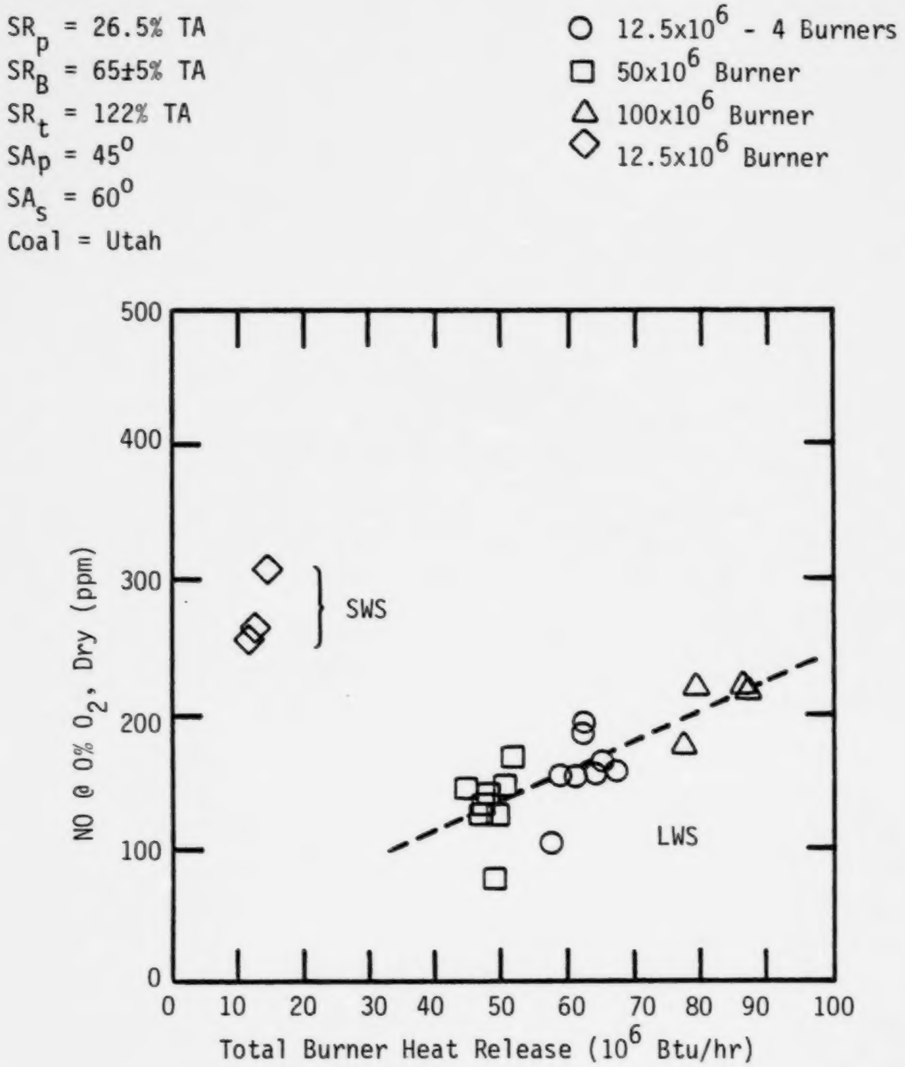
C.1.2 Long Flame DMB Designs

Excess air characteristics for the 10 and 50 x 10^6 Btu/hr burners are shown in Figure C-4. The NO dependence on excess air is similar to that presented for the short flame burners.

C.2 Fuel Composition

C.2.1 Short Flame DMB Designs

It is expected that the NO levels will be affected by fuel composition. The four fuels listed in Table C-1 were tested in the LWS using the single 100 x 10^6 Btu/hr burner and the four 12.5 x 10^6 Btu/hr burners. In addition, a series of tests were also conducted in the SWS using the single 12.5 x 10^6 Btu/hr burners. The results presented in Figure C-5 show that in the LWS furnace the Utah Nos. 1 and 2 and West Virginia coals all resulted in comparable NO levels when fired at the same overall stoichiometry. However, in the SWS



SR_p = 26.5% TA

SR_B = 65±5% TA

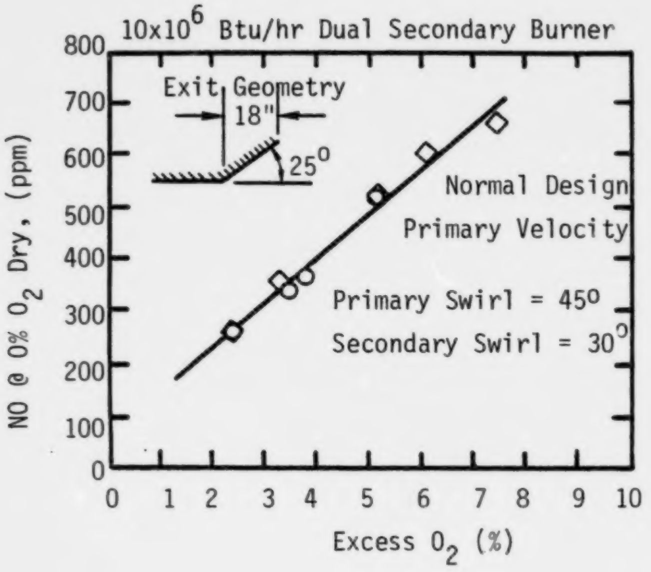
SR_t = 122% TA

SA_p = 45°

SA_s = 60°

Coal = Utah

Figure C-3. Effect of Burner Heat Release on NO Characteristics for the Short Flame DMB Designs.



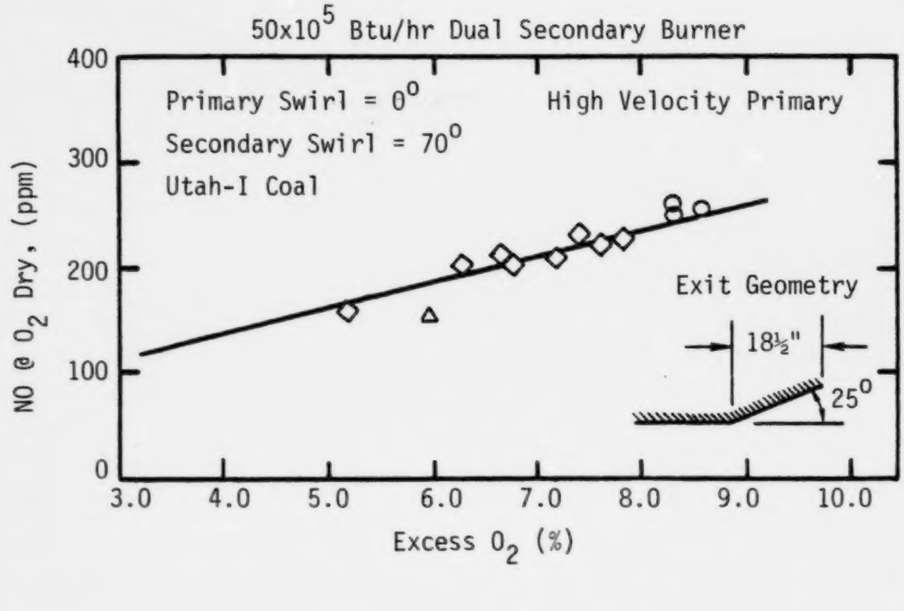


Figure C-4. Effect of Excess Air on NO Characteristics - Long Flame Burner DEsigns

Table C-1. Compositions of Coals Tested

Coal	As Received				Dry Ash Free			
Analysis	Utah I	Utah II	West Virginia	Western Kentucky	Utah I	Utah II	West Virginia	Western Kentucky
Proximate, %								
Moisture	6.39	7.41	1.29	3.68	--	--	--	--
Volatiles	38.89	38.84	31.01	38.19	45.11	46.36	36.50	48.70
Ash	7.40	8.83	13.76	17.90	--	--	--	--
Fixed Carbon	47.32	44.92	53.94	40.23	54.89	53.64	63.50	51.30
Ultimate, %								
Carbon	68.49	66.89	71.44	60.83	79.45	79.86	83.92	77.57
Hydrogen	5.20	5.33	4.82	4.47	6.03	6.36	5.66	5.70
Nitrogen	1.33	1.43	1.32	1.28	1.56	1.71	1.55	1.63
Sulfur	0.64	0.70	1.75	2.98	0.74	0.84	2.06	3.80
Ash	7.40	8.83	13.58	17.90	--	--	--	--
Oxygen	10.64	9.41	5.80	8.86	12.34	11.23	6.81	11.30
Heating Value, Btu	12,340	11,877	12,500	10,640	--	--	--	--

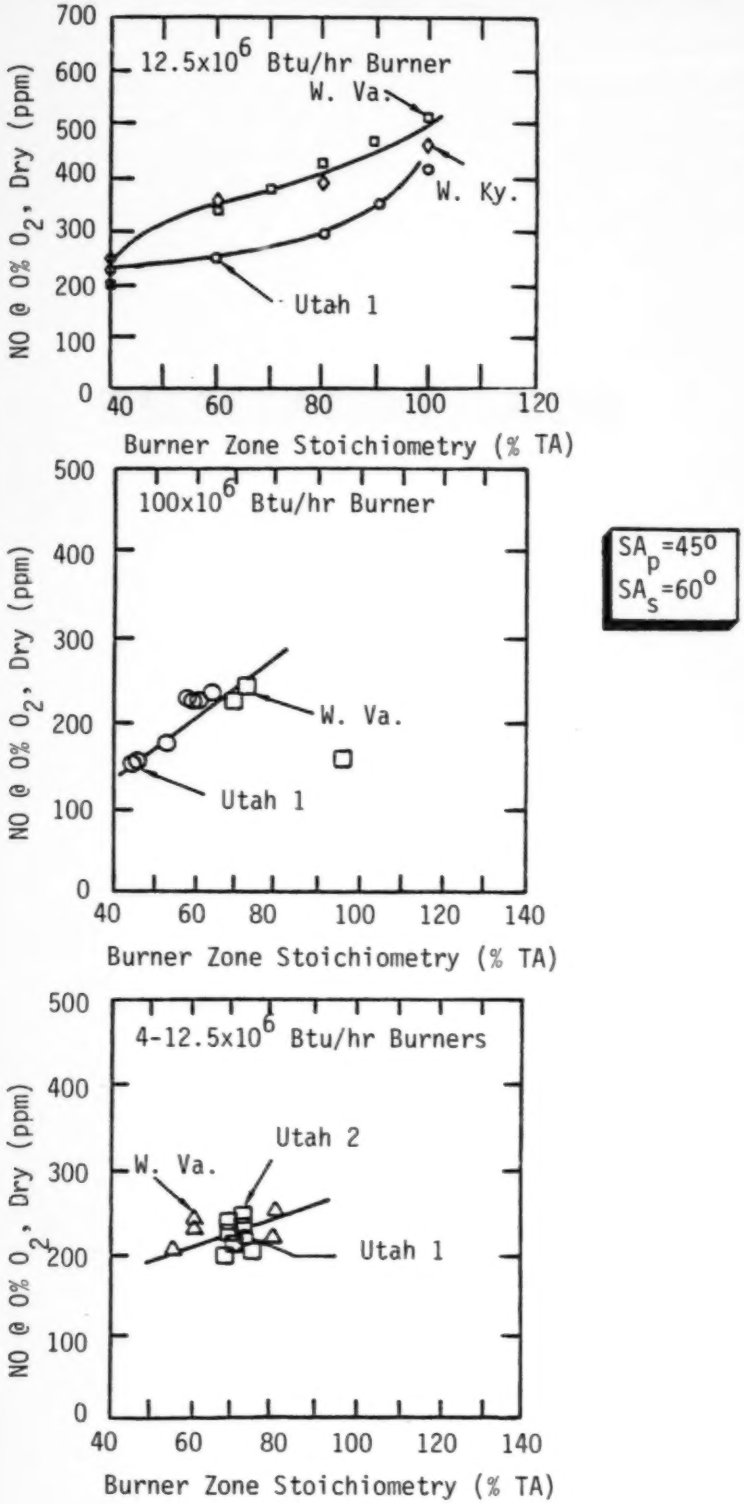


Figure C-5. NO Characteristics with Differing Fuels for Short Flame Burners.

furnace, while the West Virginia and Kentucky coals resulted in the same level of NO, the Utah No. 1 coal gave considerably lower NO values. Since these data are somewhat limited, it is not known whether the SWS fuel effects data are anomalous or the test conditions in the SWS emphasize the differences in fuel composition. Due to this uncertainty, future DMB designs should be tested with several coals to verify the effects of fuel composition.

C.2.2 Long Flame DMB Designs

The Utah and West Virginia coals were also tested in the 10 x 10^6 Btu/hr burner in the SWS. The results presented in Figure C-6 show essentially no effect of fuel composition on NO. This is consistent with the majority of the short flame burner data and suggests that NO levels are similar for coals with approximately the same composition.

C.3 DMB Full-Load Operating Limits in LWS

The overall objective of the DMB effort is to define not only the impacts of burner design and flow parameters on NO/CO, but also to establish the potential for the DMB to achieve low NO_x values approaching 0.2 lbs/10^6 Btu. While the lower limit is dependent on the coal grade and burner/boiler configuration, it is still constructive to define the full-load operating limits as a comparison against other burners evaluated in the LWS using the baseline coal.

In practice, a commercial burner is adjusted to provide the required flame shape and CO levels at full load and design excess air. The burner must then provide acceptable operation over the entire load and excess air ranges without further burner adjustments. When excess air is varied at full load, both NO and CO also vary with low excess air corresponding to low NO and high CO. This relationship between NO and CO can be plotted with excess air varying implicitly as shown in the unstaged curve in the Figure C-7.

The adjustment procedures for the DMBs tested in the research furnaces were similar to those described above, except that the adjustment of burner parameters and progressive staging allowed both CO and NO to be minimized. The improvement in performance is illustrated in the sketch above by the line joining the unstaged and staged curves. The lower staged curve represents the minimum NO for staged operation. Variation in NO and CO along this curve may be achieved by varying the burner adjustments and/or the degree of staging while maintaining the design point excess air at full load.

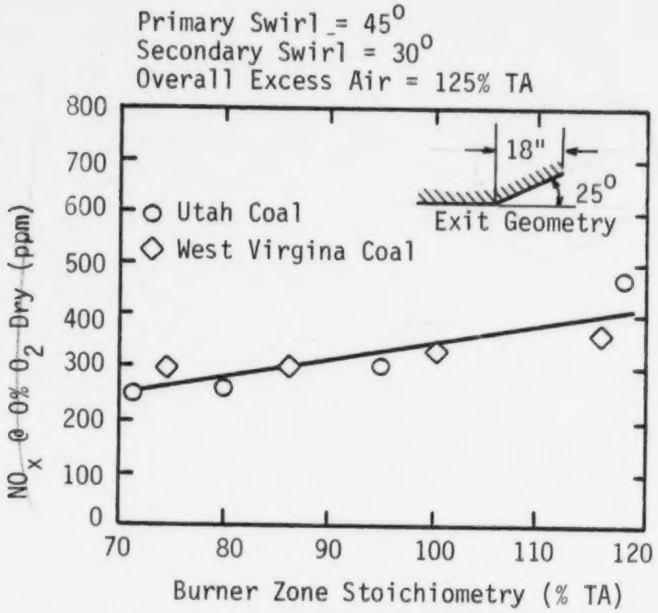


Figure C-6. Effect of Fuel Type on NO_x Emissions for the 10 x 10^6 Btu/hr Single Secondary - Long Flame Burner.

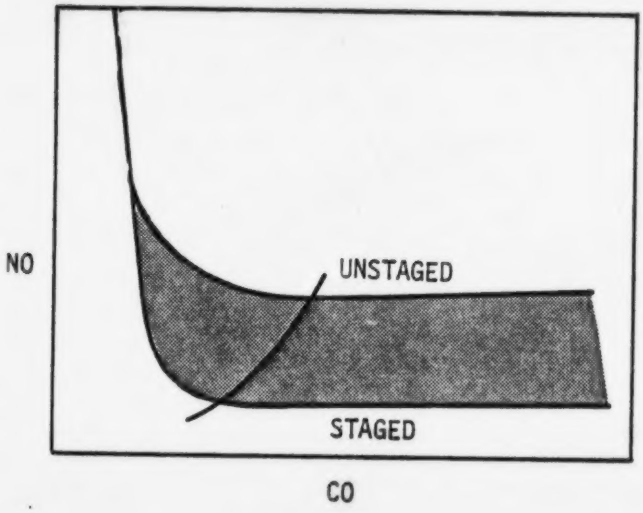
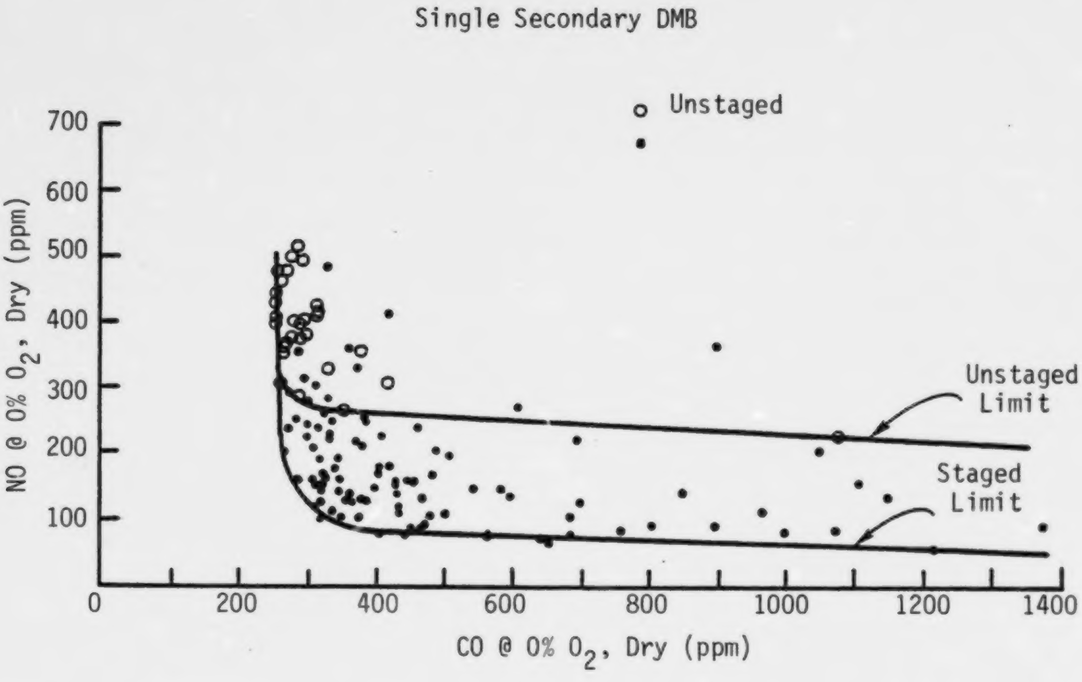


Figure C-7. Sketch of Operating Curve

The results from all full-load testing are shown on Figure C-8 for the two 50 x 10^6 burners evaluated in the LWS combustor. Note that the burner providing a long flame achieved lower values of NO than the short flame well-mixed burner. Direct comparison of the levels may, however, be unjustified because with the long flame burner some back wall flame impingement occurred. Therefore, the NO levels may be lower and CO higher than would have occurred if impingement had not resulted. The four-burner configuration and the single 100 x 10^6 Btu/hr burner results are presented in Figure C-9. These results are compared in section 4 with several commercial burners evaluated in the LWS to determine the potential for meeting the program objectives in field operating units.



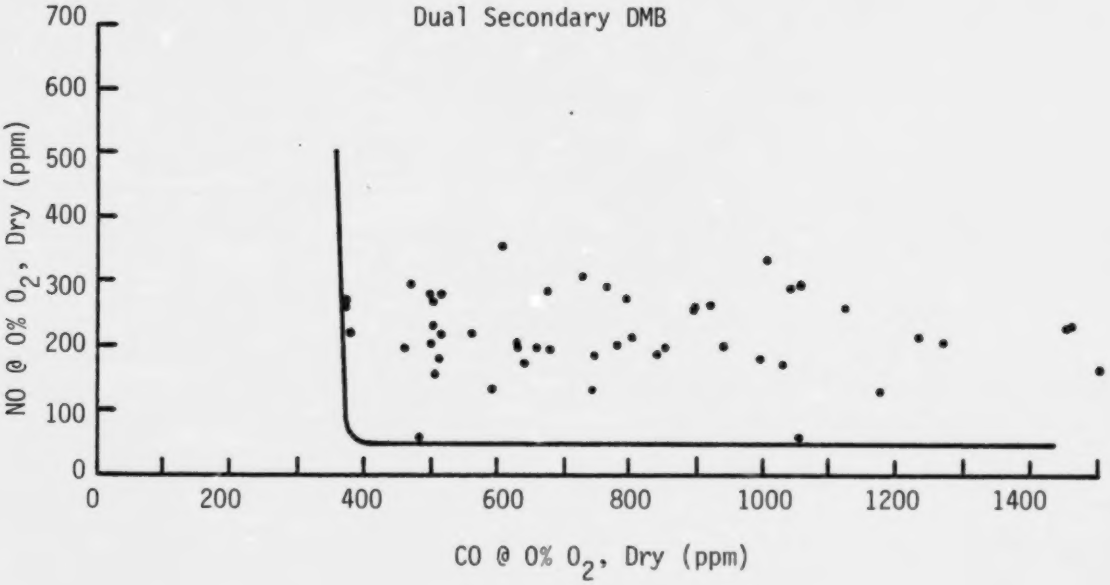


Figure C-8. Comparison of Operating Limits for Two 50×10^6 Btu/hr Burners Having Distinctly Differing Flame Characteristics.

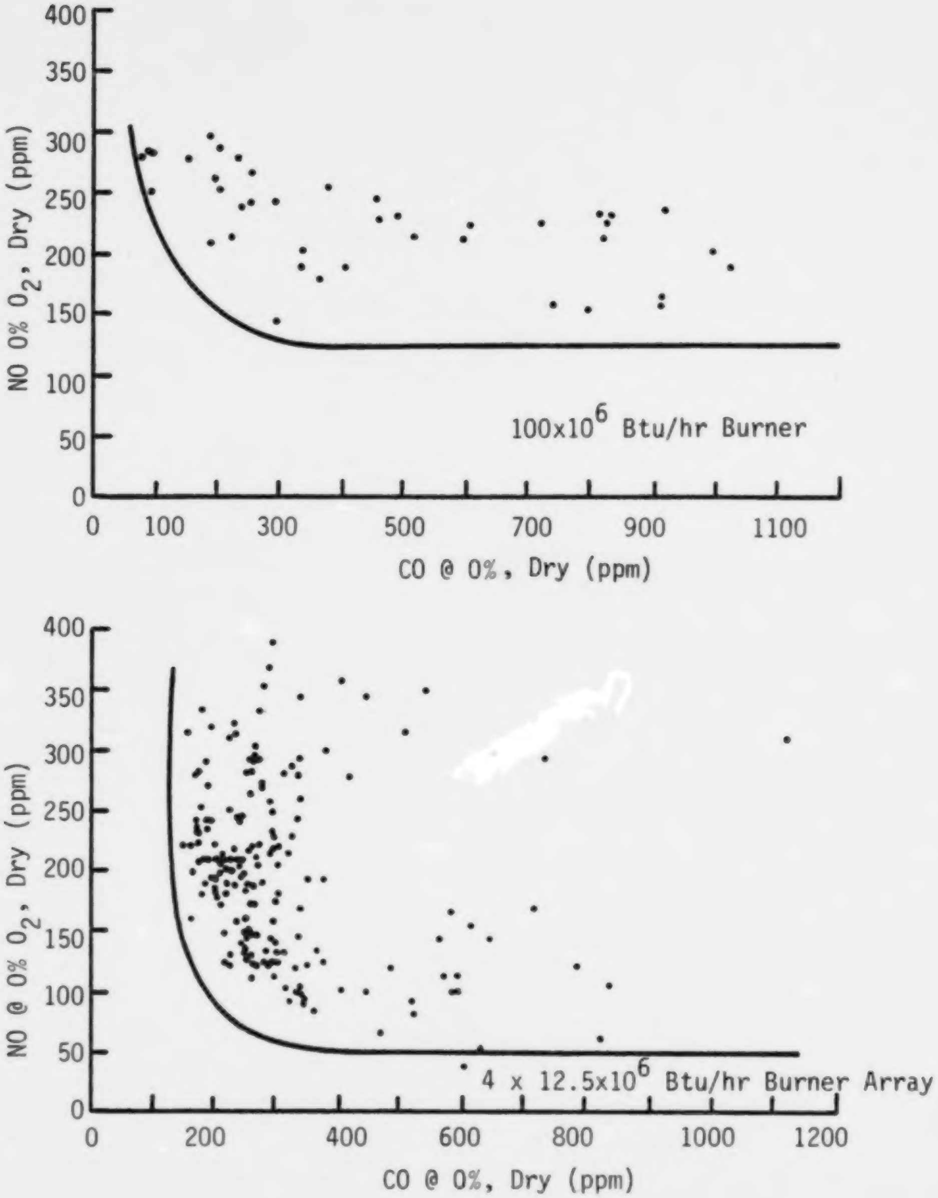


Figure C-9. Operating Limits for a Single Large Capacity Burner and a Four Burner Array of Smaller Capacity Burners.

APPENDIX D--LOW NO_x BURNER COMMERCIAL DESIGNS AND PERFORMANCE

D.1 Foster Wheeler (FW) Low NOx Burners

Based on available published data, FW low NO_x burners have been designed and tested in three configurations:

- Controlled-flow burner;
- Controlled-flow/split-flame burner, and;
- Controlled-flow/split-flame burner with variable velocity nozzle.

The controlled-flow burner design is shown in Figure D-1.

Conceptually, the burner is configured to permit only sufficient turbulence at the nozzle to maintain ignition in the near-throat region, minimize primary and secondary air mixing, and limit carbon monoxide emissions. Functionally, the burner's components and their operation are as follows:

- A perforated plate surrounds the registers and is used to measure airflow and improve burner circumferential air distribution. It also provides a discrete means for balancing air on a burner-to-burner basis;
- A dual throat with two registers splits the secondary air into two concentric streams with independently variable swirl. The mixing rate between the primary and secondary airstreams and the rate of entrainment of furnace gases can thus be varied;
- The coal nozzle used is a tapered annulus with an inner sleeve tip which is movable in the axial direction. The inner sleeve taper is different from that of the outer sleeve so that the free area of the annular opening will vary as the inner sleeve is moved. Thus, for a constant flow of primary air, the nozzle velocity can be varied to minimize NO_x emissions.

The controlled flow burner has been tested on several industrial and utility boilers. The flames were relatively short and suitable for either single-wall or opposed-wall firing.

The importance of balancing the secondary airflow between burners and the effectiveness of the FW burners in accomplishing equitable distribution is exhibited in Figure D-2. The upper part of the figure shows the concentration

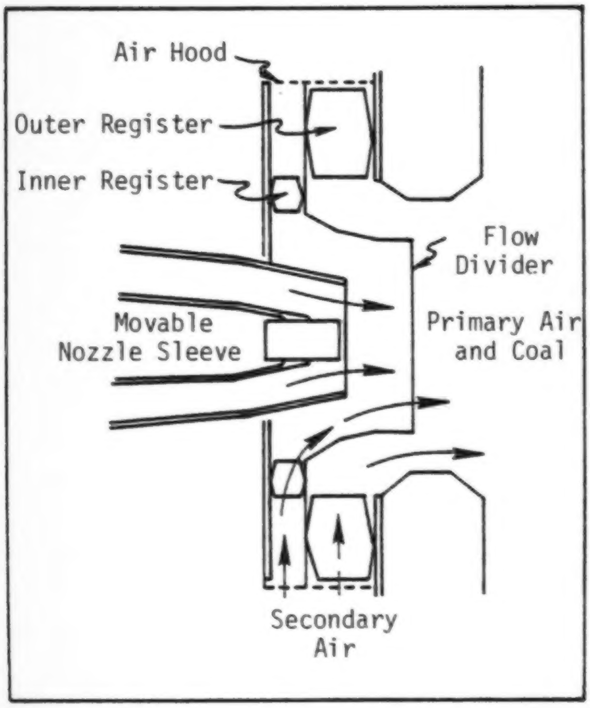
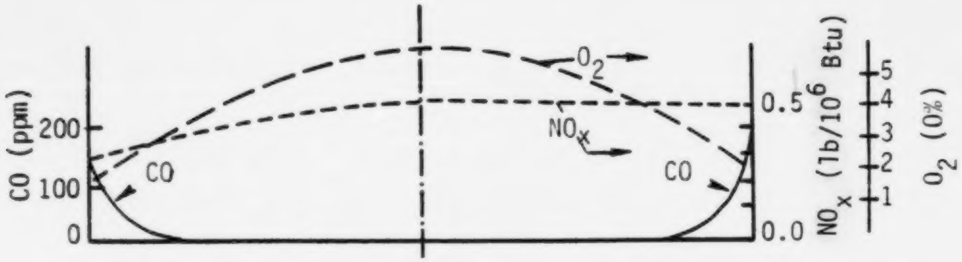

Air Hood
Outer Register
Inner Register
Flow Divider
Movable Nozzle Sleeve
Primary Air and Coal
Secondary Air

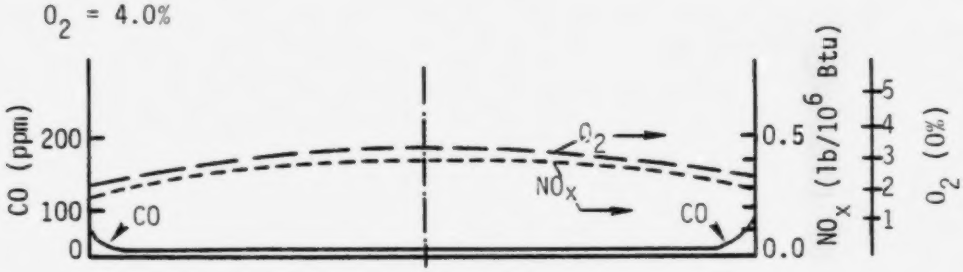
Figure D-1. Foster Wheeler Controlled Flow Burner.

NO_x = 0.55 lb/10^6 Btu
CO = 100 ppm
O_2 - 5.0%



A. Stratified Gases in Furnace

NO_x = 0.5 lb/10^6 Btu
CO = 40 ppm
O_2 = 4.0%



B. Balanced Secondary Air

Figure D-2. Effect of Secondary Air Balancing.

profiles resulting when the movable air hoods and registers of all burners were set the same. CO was highest along the sidewalls while O_2 and NO_x peaked in the center of the furnace where the CO concentration was near zero. The overall concentrations of each gas are shown to the left of the diagram. In the bottom part of the figure the concentration profiles are shown after the air hoods and registers were adjusted for equal pressure drop across the perforated plate of each burner. All of the profiles are essentially uniform and the overall levels show decreases of 9, 60, and 20 percent for NO_x, CO, and O_2, respectively, over the unbalanced secondary air conditions. These results were obtained on a 375 MW front wall fired boiler which has been retrofitted with the controlled-flow/split-flame burner.

Figure D-3 shows the results of retrofitting a four-burner (40 x 10^6 Btu/hr each) array with controlled-flow burners. The burners originally installed were high-turbulent Intervane burners, and the controlled-flow burners result in about 40 percent less NO_x at full load.

Foster Wheeler has also developed a controlled-flow/split-flame burner to achieve further NO_x reductions. The burner shown in Figure D-4 incorporates a device to split the primary air/coal flow into several distinct streams. Coal particles become concentrated within each stream and, consequently, diffuse more slowly into the secondary air. This further inhibits NO_x formation by permitting the coal to devolatilize in a more reducing zone (i.e., initial flame stoichiometry is about 70 percent TA).

Table D-1 shows the results of industrial and utility boiler retrofits with the controlled-flow and controlled-flow/split-flame burners. The controlled-flow burners reduced NO_x emissions by 50 percent in utility boilers, over the pre-NSPS burners. The addition of the split-flame coal nozzle reduced NO_x levels by 55 to 60 percent over the 35 to 40 percent achieved by the prototype controlled-flow burner. Table D-1 also shows how the burner liberation rate influences NO_x emissions and that opposed-fired boilers have lower emissions than single wall-fired boilers.

Foster Wheeler has also tested a split-flame coal nozzle which allows the primary air velocity to be varied during burner operation. This additional variable allows further optimization of fuel/air mixing rates. The fixed and variable velocity split-flame coal nozzles have been tested in a 50 x 10^6 Btu/hr single-burner test facility. The results shown in Figure D-5 indicate

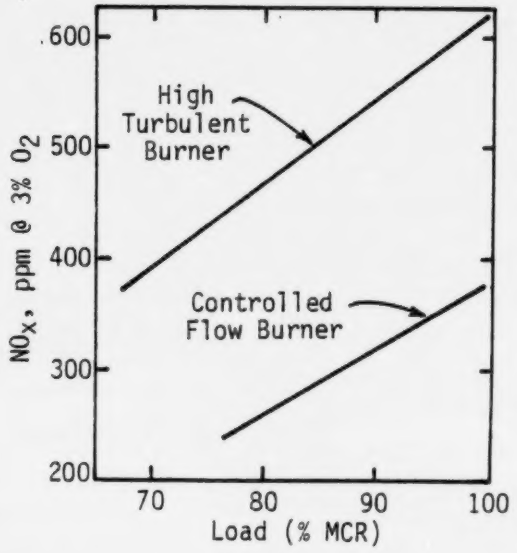


Figure D-3. Foster Wheeler Controlled Flow Burner Performance.

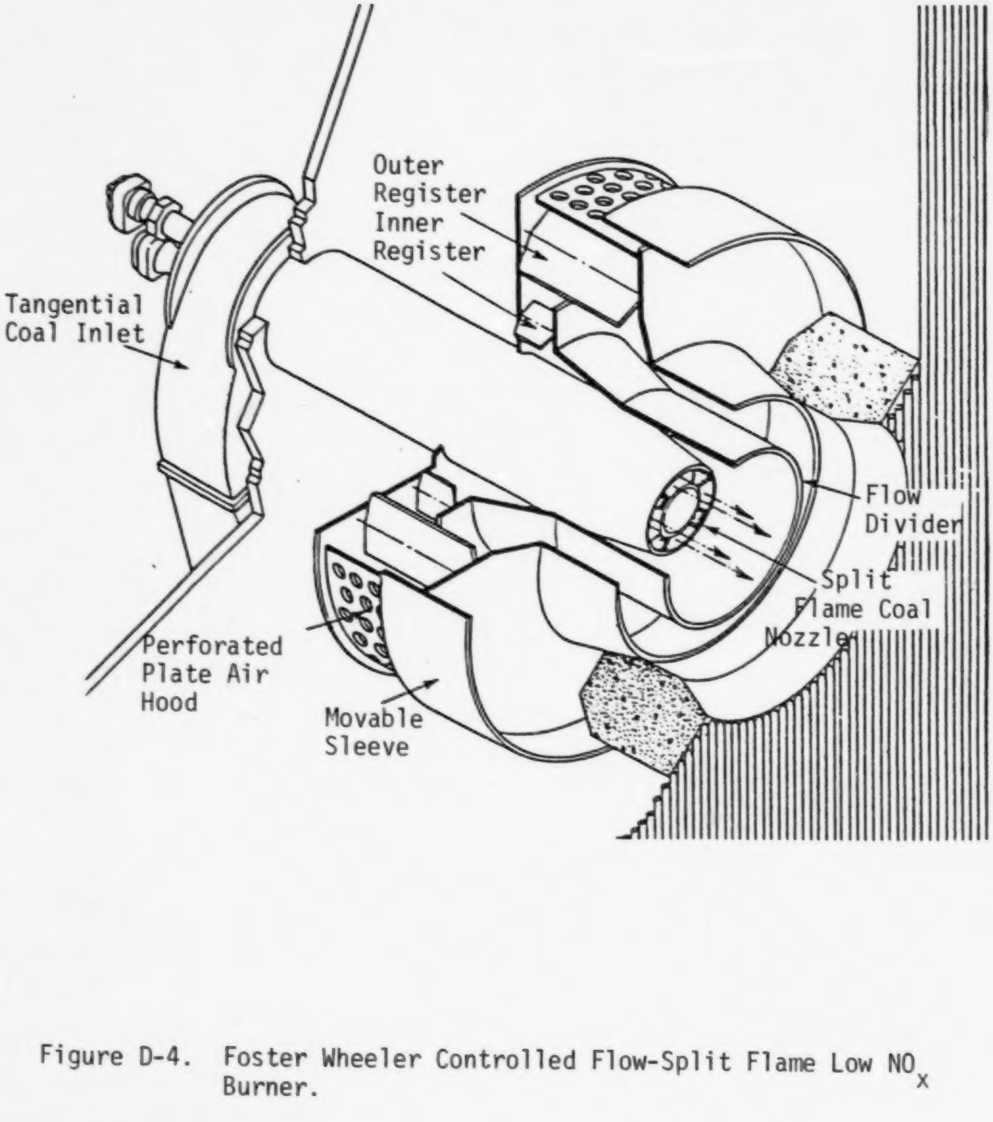

Outer Register
Inner Register
Tangential Coal Inlet
Flow Divider
Split Flame Coal Nozzle
Perforated Plate Air Hood
Movable Sleeve

Figure D-4. Foster Wheeler Controlled Flow-Split Flame Low NO_x Burner.

Table D-1. Controlled Flow and Controlled Flow-Split Flame Test Results

Burner Type	Controlled Flow - Split Flame		Controlled Flow	
Unit	Industrial	San Juan No. 1	Industrial	Japanese Utility
Unit Capacity	125,000 lb/hr	375 MW	125,000 lb/hr	265 MW
Burner Liberation Rate (10^6 Btu/hr/ft^2)	36	285	36	100
Burner Arrangement	2x2 Front Wall	4x4 Front Wall	2x2 Front Wall	2(3x4) Opposed
Uncontrolled NO_x Levels with Turbulent Burners (lb/10^6 Btu)	0.82	1.1	0.8	0.8
NO_x Reduction	60%	55%	35-40%	45-50%
Controlled NO_x Level (lb/10^6 Btu)	0.33	0.5	0.5	0.42*

* Staged 0.369/10^6 Btu

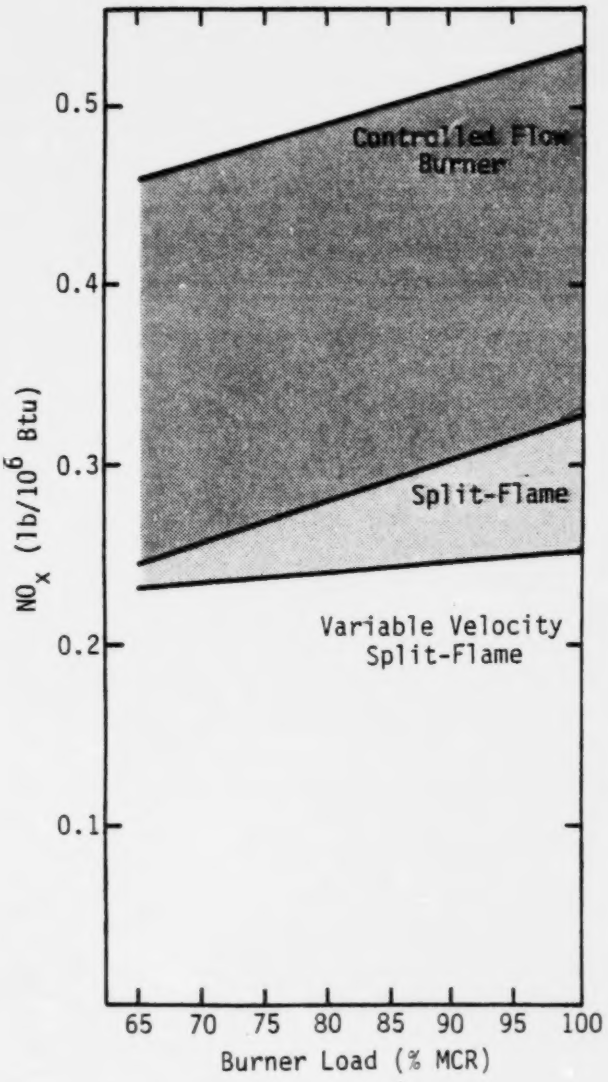


Figure D-5. Foster Wheeler Low NO_x Burner Performance in Test Facility that in comparison with the controlled-flow burner without the split flame, NO_x emissions were reduced about 35 percent with the fixed-velocity/split-flame nozzle and about 50 percent with the variable-velocity/split-flame nozzle. This is now Foster Wheeler's standard commercial burner and has been field demonstrated up to a capacity of 285×10^6 Btu/hr. Staging of this burner has further reduced NO_x to below 0.2 lb/10^6 Btu at a minimum burner stoichiometry of 96 percent. This staging was achieved with over fire air ports so that the lower portion of the furnace operated under reducing conditions. This type of operation requires low-sulfur, low-slaging coal.

D.2 Babcock and Wilcox Low NO_x Burner

The B&W low NO_x burner depicted in Figure D-6 consists of a central core through which the fuel and primary air are introduced and two annular channels accommodating the secondary air. The secondary air system features independent registers for complete primary/secondary stream mixing and the inner secondary channel contains swirl vanes to aid in stabilizing the flame. The burner is usually employed in combination with a compartmented windbox which allows for balancing the airflow between groups of burners. Two small diameter tubular passages through the swirl vanes provide access for the burner igniter and flame detector or for visual observation of the flame. Cooling of the secondary divider annulus is by airflow while the outer throat is water-cooled.

The primary nozzle is fitted with a venturi at the entrance to the burner that serves to homogenize the coal/air mixture. An adjustable plug located in the venturi throat allows fine tuning of the primary stream pressure drop so that the coal/air flow can be balanced between several burners supplied from a single pulverizer. Early tests showed that this device was prone to erosion and required frequent replacement. Babcock and Wilcox is currently testing new coal nozzle and mixing devices.

The Babcock and Wilcox low NO_x burner produces a relatively long flame. It has been field tested in seven operating boilers to date. Three of the seven fire subbituminous coals and their capacities range from 90 to 700 MW(E), while the remaining four burn bituminous coals and range in size between 250 and 665 MW(E). Figure D-7 shows the NO_x emissions from the units both with the pre-NSPS burners and after the retrofit with the low NO_x burners. The results show very little relation between unit capacity and percentage reduction of NO_x emissions, but there is a noticeable dependence on the rank of the coal fired. The average

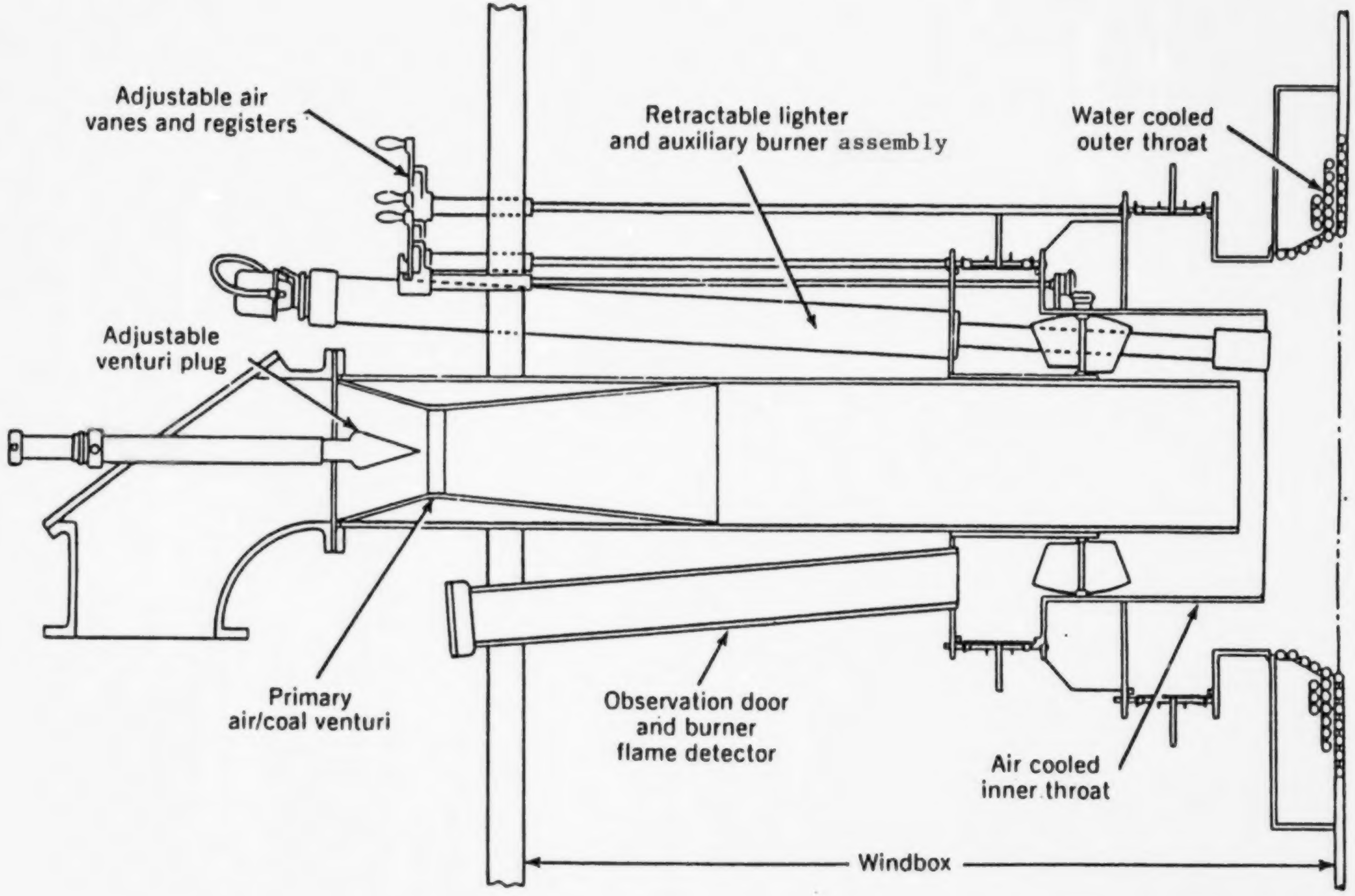

Adjustable air
vanes and registers
Retractable lighter
and auxiliary burner assembly
Water cooled
outer throat
Adjustable
venturi plug
Primary
air/coal venturi
Observation door
and burner
flame detector
Air cooled
inner throat
Windbox

Figure D-6. Babcock & Wilcox Dual Register Pulverized Coal Burner.

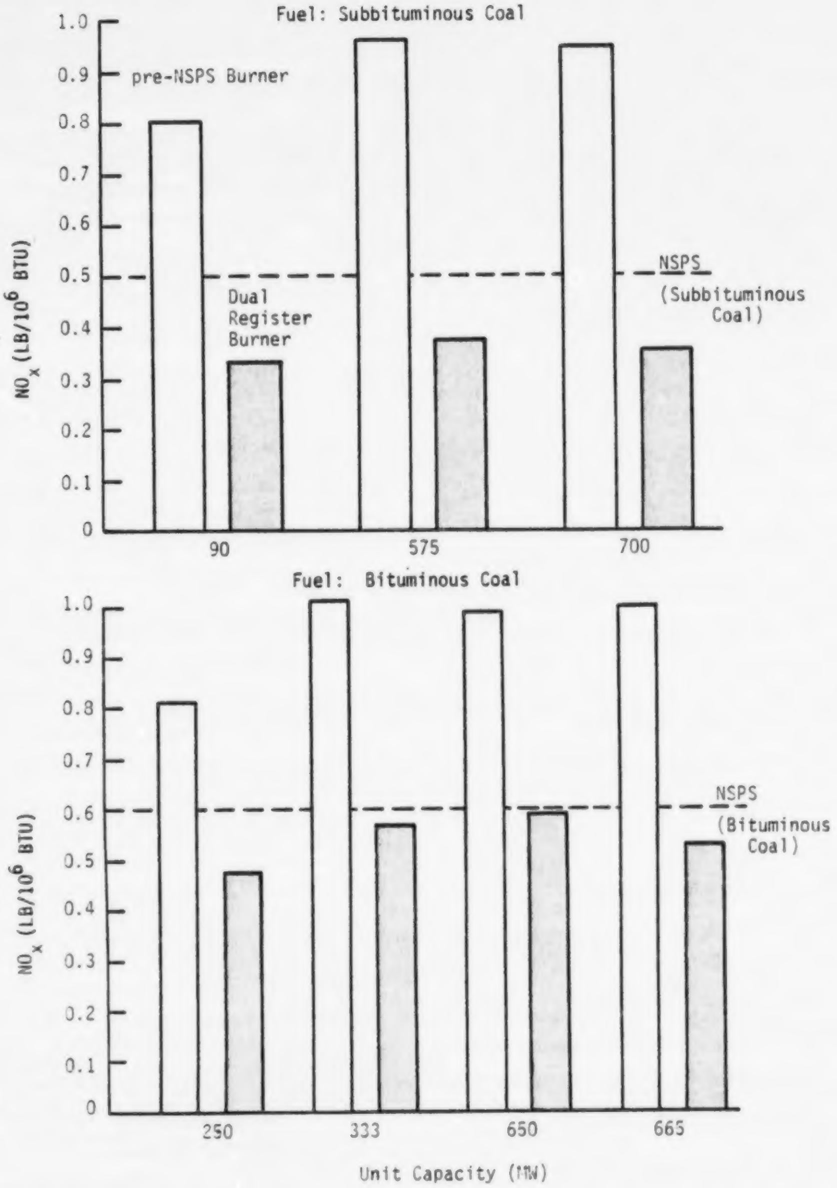


Figure D-7. B&W Dual Register Burner Field Test Results.

reduction for the bituminous-fired units was 45 percent, whereas the subbituminous units showed reductions averaging 60 percent.

APPENDIX E--COMMERCIAL BURNERS TESTED IN THE LWS

E.1 Foster Wheeler Intervane Burner (Pre-NSPS)

The Intervane burner was Foster Wheeler's standard commercial pre-NSPS pulverized coal burner and is designed to produce rapid fuel and air mixing and a stable flame over a wide range of operating conditions. The resulting flames are short and intense, and compatible with small and large furnace depths. The Intervane burner has been manufactured in a range of firing capacities up to 300 x 10^6 Btu/hr.

The burner tested in the LWS has a nominal firing capacity of 90 x 10^6 Btu/hr and is shown schematically in Figure E-1. The coal nozzle is a tapered annulus with a tangential inlet which imparts swirl to the primary coal/airstream. The coal is thrown radially outward by centrifugal force to the outer wall which has a series of "intervanes" to distribute the coal uniformly about the circumference. Secondary combustion air is supplied to the windbox and enters the burner throat through a single adjustable register. The register is normally adjusted when the burner is first installed and periodically, as required, by normal maintenance. It is not normally adjusted during load or excess air changes. The center core of the burner is also supplied with air from the windbox. This air provides cooling for the coal nozzle and is only a small fraction of the total combustion air. The burner exit has a long cylindrical portion and a short divergent portion on the exit end. In field installations the exit is formed from water-wall tubes and coated with refractory to match the desired contours.

Table E-1 lists the Intervane burner dimensions, full load operating conditions and velocities. Since the secondary air enters the throat nonaxially, the secondary area and velocity are not clearly defined. The values listed are based on axial flow in the equivalent annulus between the nozzle exit and throat and the characteristics of the Utah coal used in the previous EER burner tests.

E.2 Babcock and Wilcox Dual Register Burner (DRB) (Post-NSPS)

A schematic of the 50 x 10^6 Btu/hr Dual Register Burner (DRB) tested in the LWS is shown in Figure E-2. Hardware and operating specifications obtained from the manufacturer are included in Table E-2. The burners consist of three concentric flow channels, namely primary (coal), inner secondary and outer secondary. In commercial application combustion air to the different channels is supplied from a common windbox compartment.

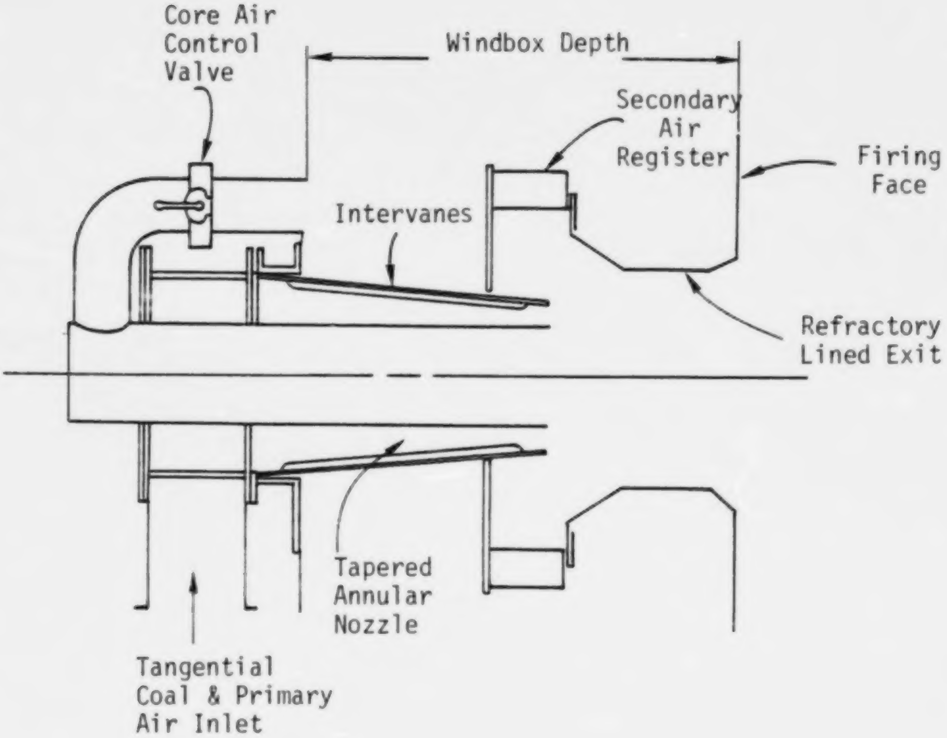

Core Air
Control
Valve
Windbox Depth
Secondary
Air
Register
Firing
Face
Intervanes
Refractory
Lined Exit
Tapered
Annular
Nozzle
Tangential
Coal & Primary
Air Inlet

Figure E-1. Foster Wheeler Intervane Burner.

Table E-1. Intervane Burner Characteristics.

Dimensions

Core Diameter	12.0
Core Area (in.^{2})	113.0
Nozzle Diameter (in.)	18.5
Primary Radial Thickness (in.)	1875
Primary Area (in.^{2})	141.1
Throat Diameter (in.)	28.0
Secondary Radial Thickness (in.)	4.75
Secondary Area (in.^{2})	346.9
Throat Area (in.^{2})	615.7
Nozzle Setback from Firing Face (in.)	24.0

Operating Conditions

Firing Capacity (10^6 Btu/hr)	90.0
Air Requirement (lb air/10^6 Btu)	7.57*
Primary Stoichiometry (% T.A.)	18
Primary Temperature (°F)	150
Overall Stoichiometry (% T.A.)	120
Windbox Temperature (°F)	600

Velocities

Primary (ft/sec)	52.2
Secondary (ft/sec)	213.7
Throat (ft/sec)	132.6

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*Based on Utah coal

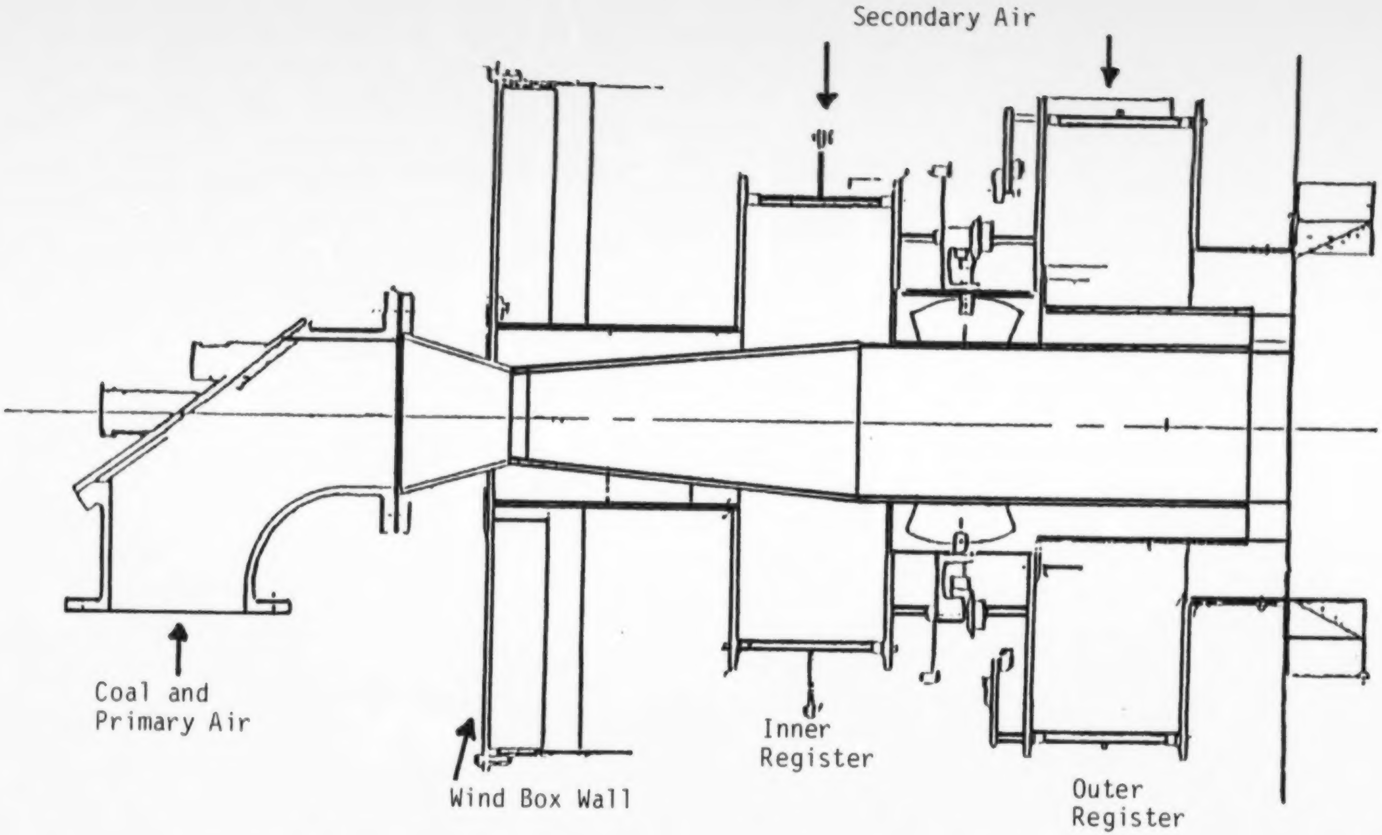
Secondary Air
Coal and
Primary Air
Wind Box Wall
Inner
Register
Outer
Register

Figure E-2. Babcock and Wilcox Burner (50×10^6 Btu/hr)

Table E-2. Babcock and Wilcox Burner Design Point Operation Specifications

Load: 50 x 10^6 Btu/hr

Stream	Operating Temperature (°F)	Mass Flow Rate (lb/min.)	Volumetric Flowrate		Velocity (ft/sec)
			(SCFM)	ACFM	
Coal	200	68	NA	NA	NA
Primary	200	136	1820	2270	50.0
Inner Secondary	600	232	3115	6230	83.5
Outer Secondary	600	406	5447	10,890	138.3

Pulverized coal is transported into the primary air pipe via a venturi-type coal distributor. The primary air pipe is 10.5 inches in diameter and terminates at a three inch recess from the burner quarl. With 20 percent of the theoretical air requirement at 50 x 10^6 Btu/hr as primary air, the injection velocity is about 50 ft/sec at 200^0F.

Inner secondary air flows into the annular channel (19.75 inches OD) around the primary pipe via a cylindrical air register with eight movable doors. The movable doors are primarily designed to control the amount of airflow, but are arranged such that the air flows in a clockwise direction when looking along the direction of flow. Inside the annular channel eight movable swirl vanes are present to produce counterclockwise tangential momentum in the secondary air. Babcock and Wilcox recommends that about 30 percent of the air be introduced into the inner secondary passage. At 600^0F this translates to about 83 ft/sec. The secondary air channel terminates at the same position as the primary air channel which is recessed three inches from the quarl section.

Outer secondary air flows through the next annular channel around the inner secondary air. A cylindrical air register with 12 movable doors (arranged in clockwise direction) is used to control the amount of air, as well as introduce some tangential momentum. This air channel terminates at the quarl and produces an air velocity of about 135 ft/sec at air temperatures of 600^0F.

The burner quarl is 5-7/16 inches long with a throat diameter of 20-1/4 inches and a half-angle of 20^0. In industrial furnaces water tubes are bent to form the quarl; however, in the LWS facility, a refractory quarl was used.

APPENDIX F--ABBREVIATIONS

Load	Burner Firing Rate - calculated as the heat input to the burner in the form of chemical energy in the fuel based on the higher heating value as received (10^6 Btu/hr).
SA_p	Swirl angle, primary - The angle of swirl vanes in the exit of the primary coal/air nozzles (degrees from axial).
SA_s	Swirl angle, secondary - The angle of swirl vanes in the annular secondary air passage (degrees from axial).
SR_B	Burner zone stoichiometry - The air passing through the burner throat including the primary and secondary air passages as a fraction of the air required for complete combustion (percent of theoretical air).
SR_p	Primary stoichiometry - The air present in the primary stream as a fraction of the air required for complete combustion (percent of theoretical air).
SR_T	Overall stoichiometry - The total air supplied to the burner as a percentage of the air required for complete combustion (percent of theoretical air).
T_p	Primary air temperature (^oF).
T_s	Secondary or tertiary air temperature (^oF).

APPENDIX G - METRIC CONVERSIONS

1. One Pound Per Hour = 0.455 kg Per Hour
2. 10^6 Btu Per Hour = 0.293 Thermal MW
3. 1 Ft = 0.32 M
4. $(°F - 32)\frac{5}{9} = °C$
5. 0.1 Lb NO_x (ASNO_2) Per 10^6Btu = 43 Nanogram Per Joule
6. 1 Inch = 2.54 cm
7. 1 Btu/lb = 0.43 Joule Per Gram
8. 1 Standard Ft^3 Per Minute = 0.0283 m^3 Per Min